



FACILITY FORM 602

N70-31901

(ACCESSION NUMBER)

(THRU)

(PAGES)

CR-110505

(NASA CR OR TMX OR AD NUMBER)

(CODE)

03

(CATEGORY)

TEXAS INSTRUMENTS
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HEAT STERILIZABLE AND IMPACT RESISTANT
Ni-Cd BATTERY DEVELOPMENT

SECTION III.
TASK IX IMPACT TESTING SECTION

Jet Propulsion Laboratory
Contract No. 951972, Modification No. 14

Final Report, Volume III
January 1969 -- January 1970

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This work was performed for the Jet Propulsion Laboratory,
California Institute of Technology, sponsored by the National
Aeronautics and Space Administration under Contract NAS-7-100;
Task Order No. RD-26.

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SECTION III.

TASK IV IMPACT TESTING SECTION

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SECTION III.

IMPACT RESISTANT COMPONENTS AND CELL DEVELOPMENT

A. DEVELOPMENT OF HIGH-IMPACT TESTING FACILITY

1. Introduction

In support of the unmanned exploration of the planets, the Jet Propulsion Laboratory has been actively interested in the development of electronic equipment capable of hard landing survival since 1959. Of immediate concern was the development of a battery power supply required for a proposed hard landing on Mars in 1973. Based upon entry conditions calculated from information about the gravitational and atmospheric conditions on Mars, a design goal has been selected by JPL for all equipment aboard this hard landing capsule. At present, a shock level of approximately 4000 g (square pulse for 1 msec) from an impact velocity of 120 ft/sec is anticipated as the kind of environment to which capsule components will be subjected. Texas Instruments, as a developer of heat-sterilizable Ni-Cd space batteries, has long been interested in the cell's resistance to shock and vibration environments. During 1969, over 100 high-capacity cells (25 AH) capable of heat sterilization were manufactured for evaluation. Under the same contract with JPL, development work concerned with prime cell components was conducted to further extend the knowledge required for high-impact cell design. In order to carry out this devel-



opment work and the evaluation of complete cells, it was necessary to simulate the anticipated shock environment. The scope of this section is to describe in detail the existing impact facility at Texas Instruments. We wish to describe its development and the resultant capabilities at our disposal.

2. Mechanical Design of Testing Equipment

a) General Description:

In the design of the test facility, prime concern was given to the means of achievement of the specified shock pulse (the desired shape, magnitude and length). Secondary concerns were the great number of cell configurations (shapes and sizes) that must be accommodated. Finally, a facility with enough versatility to permit the variation of the shock pulse was needed for a complete study of the dynamic response of components. The testing method selected to meet these demands consists of the following arrangement: The specimen is mounted within a sliding carriage by means of adjustable fixturing. Carriage and specimen are then accelerated to the desired velocity. The sliding carriage is finally impacted against an expendable target and massive anvil to produce the desired shock pulse. The overall system is shown in Figure 1.

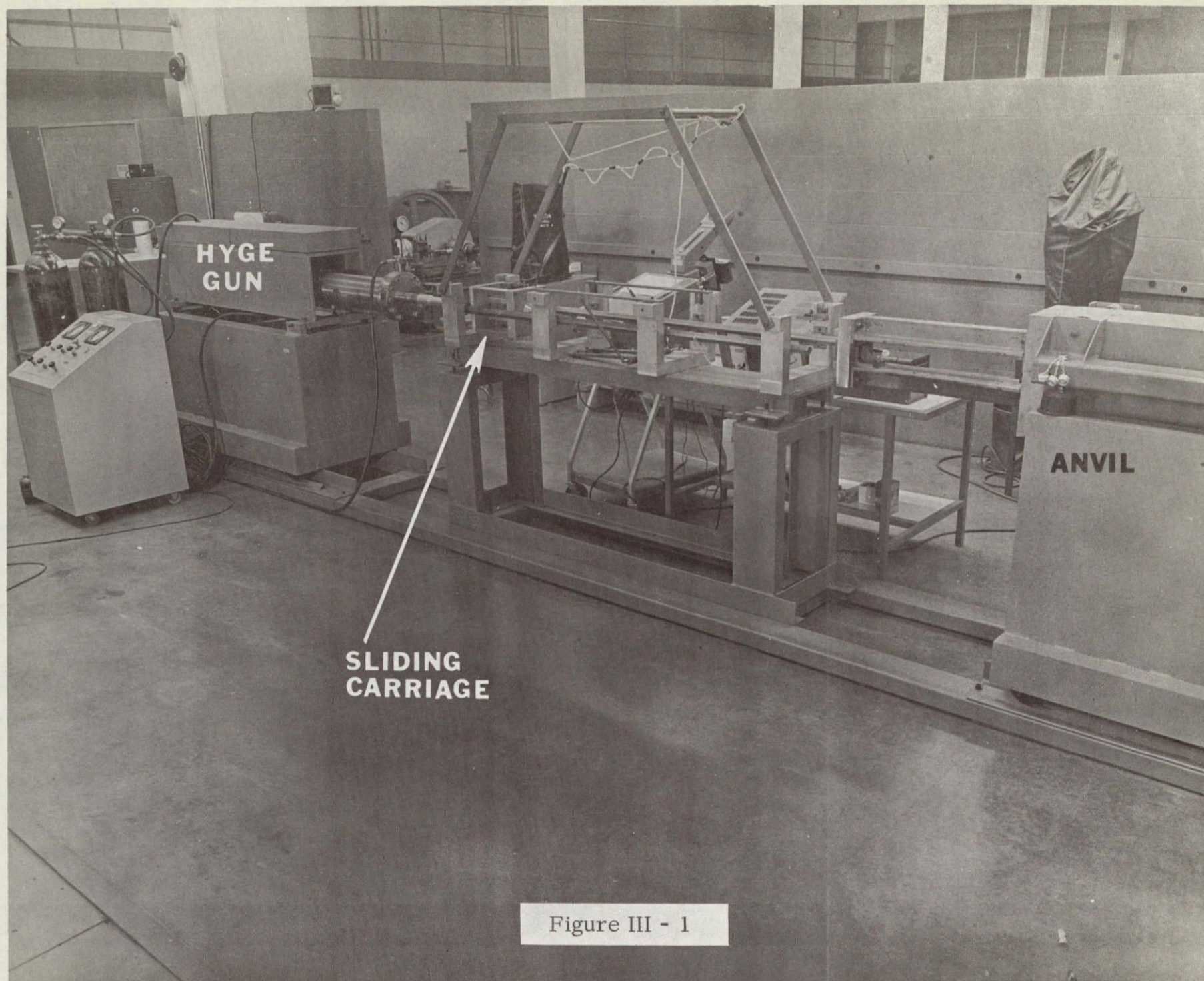


Figure III - 1



b) HYGE Gun:

The accelerator is a commercial unit produced by the Consolidated Electro-Dynamics Corp., and called a HYGE gun. It is essentially a pneumatic cylinder with the piston operated by a pressure differential. The gun used at present has a 6" dia. bore with a piston area ratio of 5.2:1 and compressed N_2 is the operating fluid. Thus with approximately 2000 psi N_2 supply this unit is rated at 40,000 lbs. thrust and capable of accelerating a 40 lb. mass to a velocity of 125 ft./sec. In order to gain more complete firing control than provided by automatic unbalance of equilibrium, we have added an auxiliary firing valve. Repeatability of ± 1 ft./sec. is attainable on a given sequence of testing using the manual firing valve. Oscillograph traces of acceleration - time were recorded to examine the carriage input.

c) Carriage & Specimen:

The sliding carriage mass is approximately 32.5 lbs. All structural members were machined from 7075-T651 aluminum alloy plate. The carriage itself sustains repeated central impacts during acceleration and more importantly at impact. Of prime interest is the versatile fixture arrangement. We are able to accommodate various cell configurations and sizes in any orientation.



Changes between tests involve very little time and ensure a more rapid evaluation of particular cells or components. Potting material has been unnecessary with the obvious exception of impact at the terminal end of a cell. This ensures a true evaluation of the cell in the absence of most shock attenuating material. The impact end of the carriage is fitted with a hardened steel (M2) impact tool. Several penetrater diameters have been ground and the design has proven quite satisfactory in resisting the repeated impact energy of over 7000 ft.-lbs. As will be shown, the tool diameter is the prime variable affecting the g-level attained at impact and the ability to change this tool readily is of great convenience.

d) Anvil:

The anvil, against which the sliding carriage impacts, is shown on the extreme right of Figure 1. It is basically a 2000 lb block of steel mounted on a steel framework that is free to roll on tracks. At full impact conditions, the entire mass (approx. 2500 lbs) will move 2-3 inches while shock absorbers arrest its motion and return it to the original position. A structural steel extension is fastened to the main anvil block and serves to position the main copper target block. This copper block is our means of achieving the desired uniform



deceleration of the carriage mass as we plastically deform the copper. Typically, the target is annealed copper (#110) 3" x 3" x 1-1/2" thick.

3. Instrumentation System

a) General Description:

In order to fully evaluate a cell during shock environments, it is desirable to monitor the cell's voltage during impact and an acceleration-time history for a full description of the event. The acquisition of data from a projectile experiencing this kind of motion is not straightforward. The lead wire arrangement found most satisfactory is a combination of high strength steel slide-wire and the two-point suspension loop of 1/4 inch dia. elastic shock cord (bungee). Combined with an abrasion resistant harness, this system provides adequate progressive restraint to control the lead wire motion.

b) Cell Voltage:

Ni-Cd cells present a nominal 1.3 volt open circuit signal. A signal of this magnitude presents no particular measurement problem once the lead wire motion has been controlled. Thus, we are able to display this dc voltage level directly on the upper beam of the oscilloscope. Hence, a cell which is truly impact resistant



will exhibit a constant potential (straight line at dc level) throughout the impact period. Complete failure at impact (plates and tab fracture or complete shorting) will exhibit a drop to a lower dc level line. In addition we are able to observe any momentary shorts that may occur within the cell only during the period of impact. Should it be desirable, an external load is easily placed across the cell and thus discharge voltage (or charge voltage) can be measured during the actual impact.

c) Velocity:

The velocity of the carriage at impact is accomplished essentially by measuring the actual time of travel between two points of known distance apart. The moving carriage momentarily completes two circuits during its travel and the two voltage pulses are used to start and stop an electronic counter which records the time interval (μ sec) between the known distance. Although basically very simple, this technique is highly reliable and possesses several advantages over other methods. Accuracy is better than 0.5 ft/sec.

The velocity measurement itself is made only one inch before the point of impact. This system readily supplies an equally reliable trigger pulse for the



oscilloscope as shown in Figure III-2.

d) Acceleration:

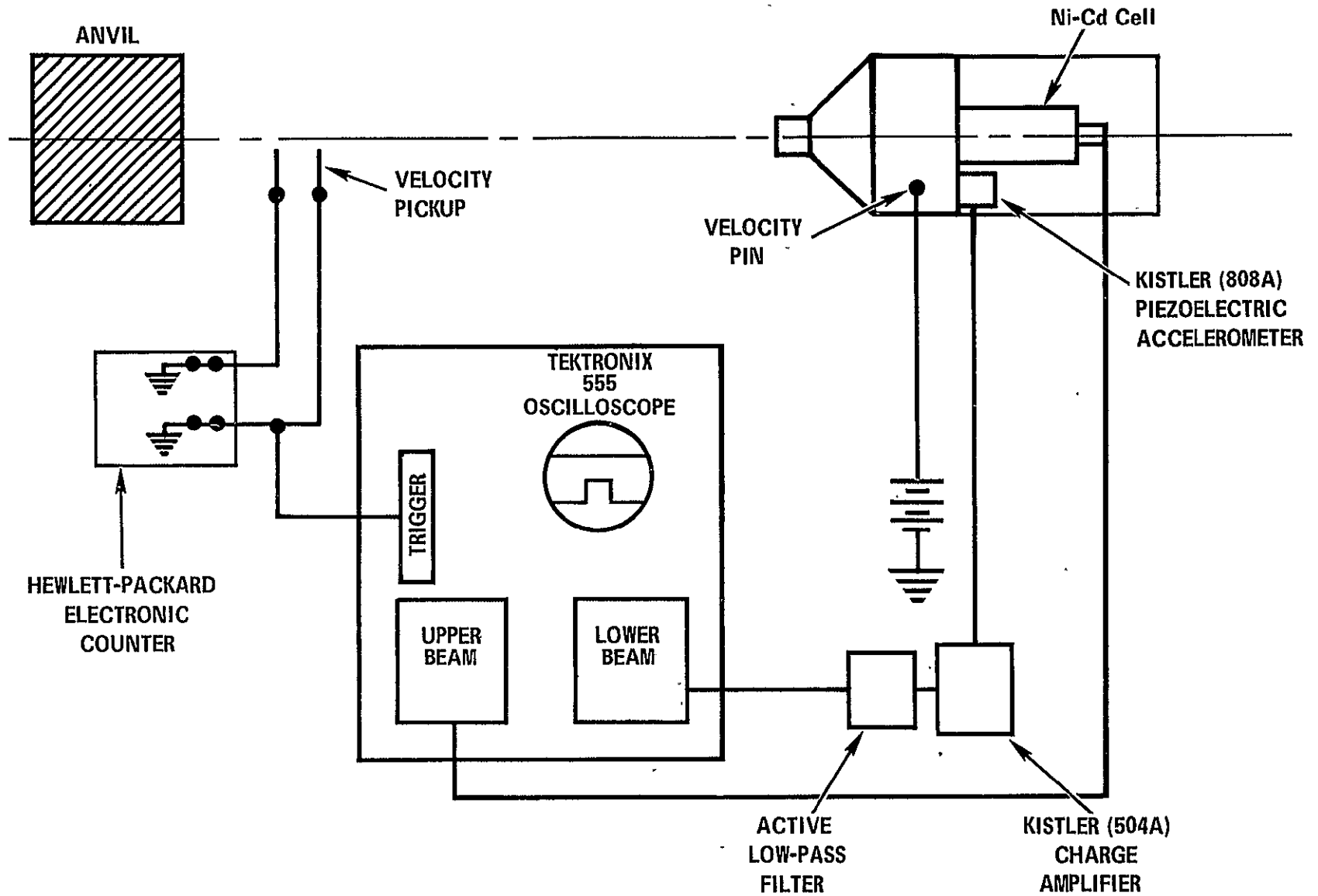
(1) Instrumentation:

The measurement of the shock pulse associated with the deceleration of the carriage and specimen is obtained by a piezoelectric accelerometer mounted on the main carriage plate. The high impedance charge output of the transducer (Kistler 808A or 805A) is converted by an electrostatic charge amplifier (Kistler 504A) to the low impedance voltage signal desired for display on an oscilloscope (Tektronix 555). A passive transducer resonance filter is also provided at the amplifier. With this conditioning, the signal is connected to a Tektronix 1A7 plug-in at the scope.

(2) Low Frequency Response:

Good low-frequency response of the system is a necessity in order to follow the long duration of the rectangular pulse created. In the present instrumentation system, the dominant time constant is determined by the product of the feedback capacitor and resistor within the charge amplifier. With the present instrument settings we have a time constant of 5.0 seconds ($< 0.1\text{HZ}$) which is nearly

Figure III - 2





100 times better than the requirement for 2% accuracy.

(3) High-Frequency Response:

A Fourier spectrum analysis of a rectangular pulse shows that most of the energy of that pulse is contained in the frequency band from zero to $1/T$ (T =pulse length). It has also been shown that in order to prevent excessive rounding of corners, $f_H \geq 10/T$. In the present case ($T=.001$ sec) it was felt that $f_H=10\text{KHZ}$ would be quite adequate to accommodate the rectangular pulse and rapid rise time. The high frequency response of the instrumentation is theoretically limited by the first mechanical resonance of the accelerometer (mass-spring) itself. Manufacturers generally trust approximately $f_n/5$ as the useable ($\pm 5\%$) frequency range and this implies approximately 8KHZ for our system. This value approached the desired 10KHZ and we expected fairly good performance. The rise time of our pulse is approximately 30 μsec and the system seems to have more than adequate high-frequency response to follow this pulse.

(4) Effect of Vibrations:

It was felt that lower frequency vibration might be a disturbing influence and we calculated the



natural frequencies of key structural elements within the carriage for several modes of vibration. Later in the testing program, one rather important design or structural modification was made to the sliding carriage which significantly improved the overall performance. The original $8\frac{1}{2}$ " x $8\frac{1}{2}$ " x 2" thick aluminum main plate was replaced by a conical block of the same aluminum alloy. The shape is best described as a conical frustrum with a square base ($8\frac{1}{2}$ " x $8\frac{1}{2}$ ") and a circular top (4" dia.). The conical included angle is 80° providing a 6" height or "thickness" replacing the original 2" thick plate. This angle is the same as the tool angle and the 4" dia. top matches the tool's base exactly. Thus, we have a smoother transition from tool to specimen mounting and have essentially eliminated the vibration problems that previously existed. Because of the increased weight and associated momentum, tools of larger diameters were required to maintain the upper g-levels. However, the resultant shift in calibration was fully compensated by the increased structural rigidity and the resultant improvement of system performance.



(5) Filtration & Calibration:

Alternatively, such vibration and the frequencies involved were best eliminated from the system by electronic filtration. Thus, an active low-pass filter network was designed and employed to attenuate all but the fundamental vibration frequency.

4. Overall System Analysis

Oscillographs were taken during the development phase and indicated the g-levels and pulse lengths typically achieved. They were used as the basis for calculations to check the overall system accuracy. Our present collision into copper closely approximates the perfectly plastic collision yet we experience some rebound. Considering the applicable relations:

Impulse = Δ Momentum:

$$\int_{t_0}^t F dt = m (V_{m2} - V_{m1}) \quad (1)$$

Conservation of Momentum:

$$m V_{m1} + \cancel{M V_{M1}} = m V_{m2} + M V_{M2} \quad (2)$$

Conservation of Total System Energy:

$$\begin{aligned} KE_{m1} + KE_{M1} &= KE_{m2} + KE_{M2} \\ + E \text{ Absorbed in} &+ \cancel{E \text{ Heat Light.}} \\ \text{Deforming Copper} &\text{Sound} \end{aligned} \quad (3)$$



Equation (1) may be solved for V_{m2} by using the measured V_{m1} and the area of the acceleration-time trace. This would permit the solution of equation (2) for V_{M2} directly assuming m and M were known. Finally equation (3) could be solved for the force required to deform the copper since the depth of penetration is measured. Thus we can compare this calculated force to that force associated with the test's mean g-level. The results are given in Table III A-1.

In the tabulated calculations we have noted the very small anvil velocity that is implied (ref. col. V_{M2}) and the fairly low carriage rebound velocity (ref. col. V_{m2}) that follows the collision and thereby established the system's proximity to the classical plastic case. Late in the testing program high speed photographic analysis was also used to investigate the overall system.

5. System Calibration

During the development phase and the initial testing program, a large amount of data was generated by this testing facility. Many separate studies were employed to extend the typical range of values and the total of this information constitutes the overall system calibration. The blunt-end cylindrical tools are responsible for achieving a square deceleration pulse with excellent

TABLE III A-1

Tool Dia. (in)	Meas. Impact Velocity V_{m1} (ft/sec)	Calc. Rebound Velocity V_{m2} (ft/sec)	Calc. Velocity V_{m2} (ft/sec)	(1)	(2)	(3)	(4)	(5)
				F_{mean} Calc. from full System Analysis as Described Above	$F_{mean}=ma_{mean}$ a_{mean} Meas. from Photo	$F_{mean}=ma_{mean}$ from uniform deceleration $a_{mean} = \frac{(V_{m1})^2}{2S}$	F_{mean} Im- pulse Time %	$F_{pk}=ma_{pk}$ a_{pk} Measured From Photo
5/8	82	4.5	1.1	56,800#	55,200#	57,800#	51,400#	65,000#
3/4	91	0	1.1	78,700#	74,800#	79,700#	68,800#	87,700
7/8	108.5	3.8	1.5	104,000#	104,000#	105,000#	103,000#	122,000#
1	115.5	10.8	1.6	135,000#	119,000#	138,000#	110,000#	140,000#



rise times. The actual g-level and pulse length, however, remain flexible and are under the control of the operator within certain bounds. It is this capability (calibration) that is best shown in Figures III-3 and III-4 which illustrate the effects of changing the main system variables.

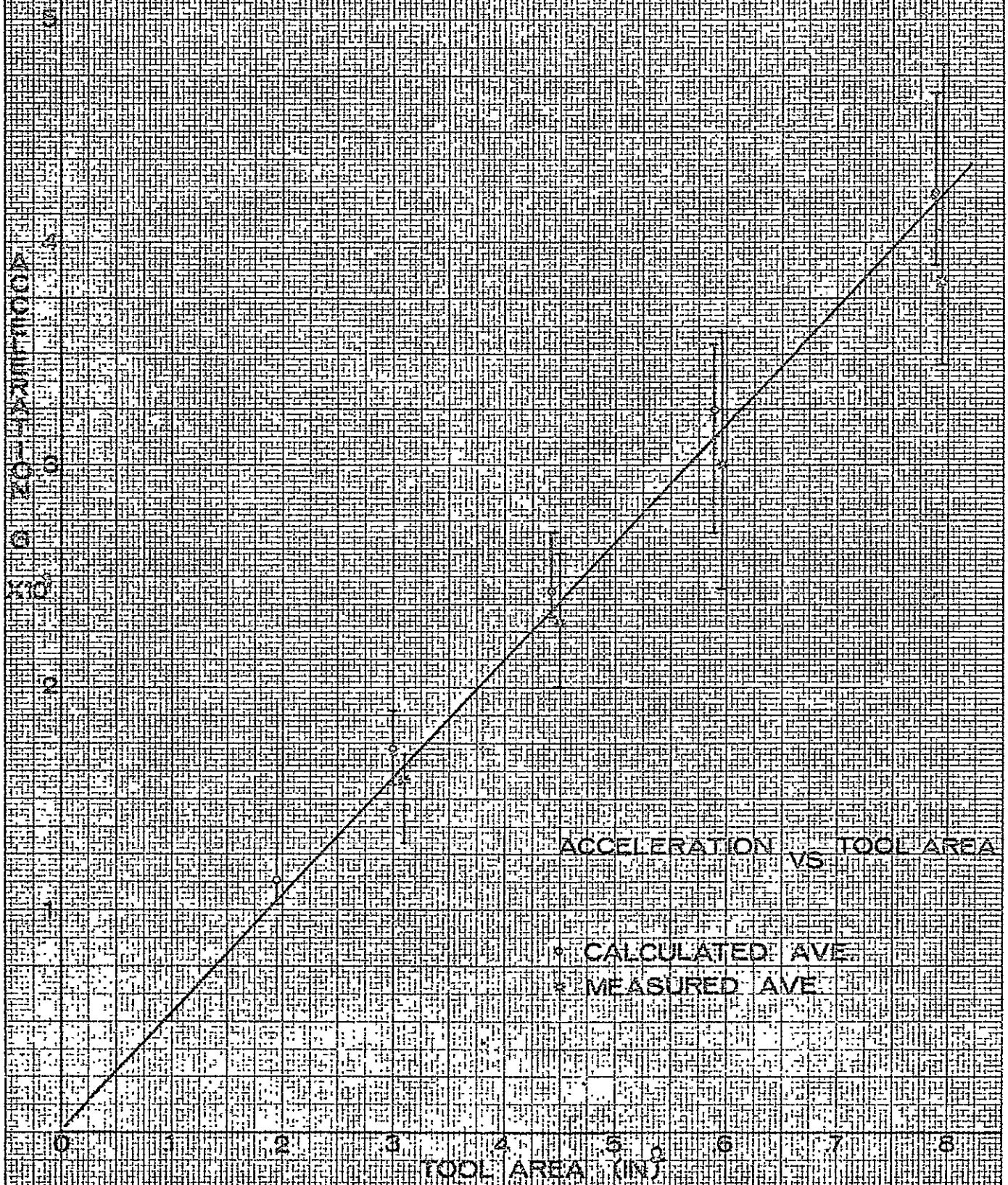
a) g-Level vs Tool Diameter

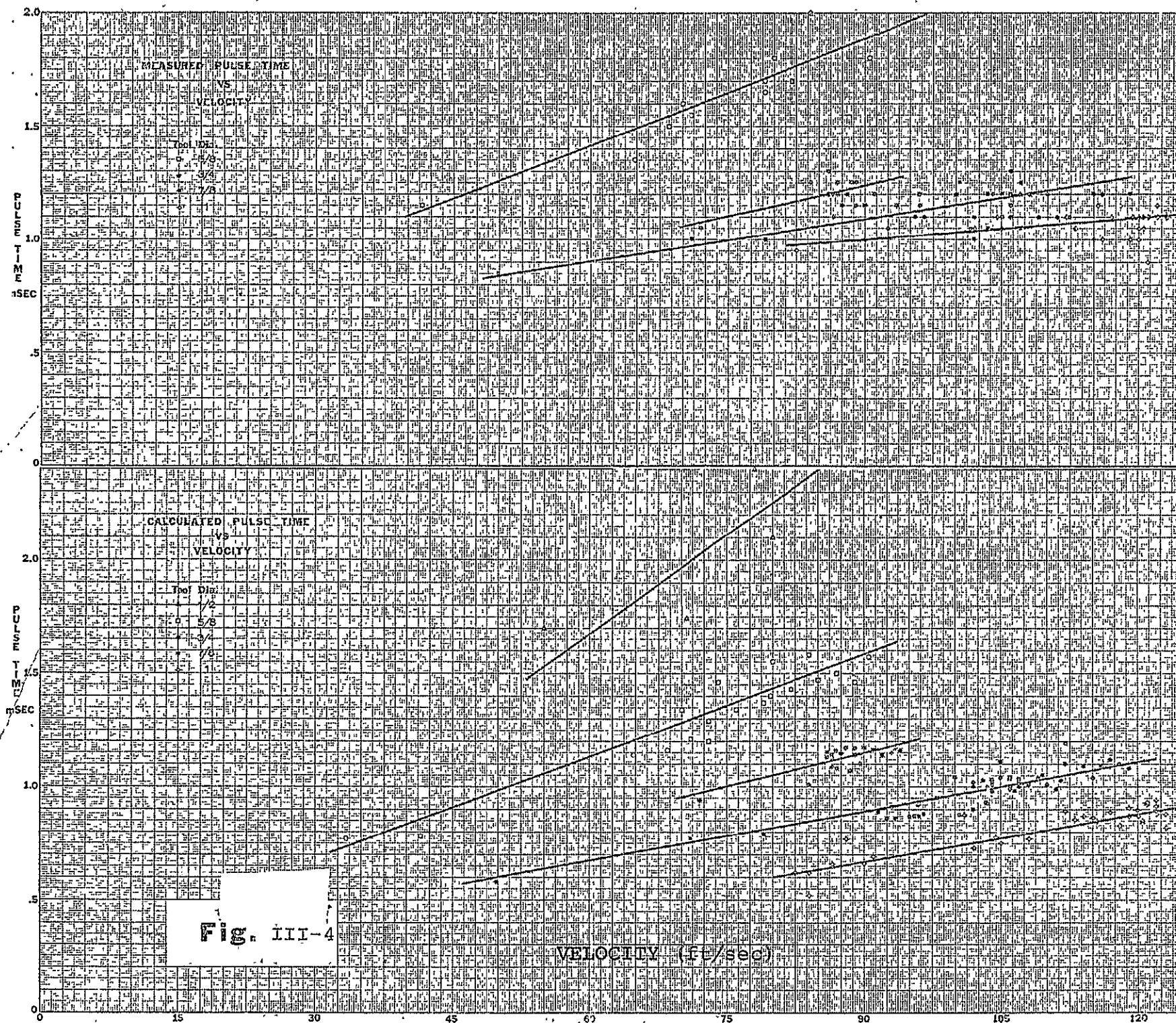
As indicated earlier, tool diameter or the area of the penetrator is the prime variable affecting g-level. Assuming a fairly constant state of three-dimensional stress during copper deformation, the total force (g-level) would be directly proportional to the area. Figure III-3 shows this linear relationship as exhibited over many impact tests.

b) g-Level vs Velocity

The carriage mass, equipped with any given tool diameter may be fired at any desired velocity as regulated by the selection of differential pressure. The practical limits are: 1) low velocity provides a short (time) pulse length which was not of interest in this study, 2) high velocity provides a very deep penetration (tool may be embedded in the copper) and if a very long (time) pulse length is not required, it is avoided as a matter of convenience.

PRELIMINARY IMPACT LEVEL CALIBRATION







The point is that in saying the g-level is primarily a function of tool diameter, we would expect to see a g-level (for any given tool) that is invariant with respect to velocity. We have varied the test velocity for each tool and illustrated the relative invariance of g-level.

c) Pulse Length vs Velocity

With a constant carriage mass we would expect that, as a consequence of the above result, the pulse length would be linearly related to the test velocity ($\int F dt = MAV$). Figure III-4 illustrates this result. The lines drawn in Figure III-4 indicate the degree of sensitivity (pulse length as a function of velocity) for each tool. Each tool has a different sensitivity as we have shown it in Figure III-4 but this is simply a result of including an actual area function (d^2) within our third parameter tool diameter (d).

6. Summary

We have developed a versatile impact testing machine capable of evaluating most cell configurations within the range of 1000g - 4800g and 0.8 - 2.0 msec. This has been accomplished by changing only two variables -- tool diameter and carriage velocity. The ability to cover



such a range this easily is of great convenience during a given testing program. However, this in no way represents the limit ranges for this machine. Two other prime variables (carriage mass and target materials) have been held constant throughout the system development described in this report. If desirable to study the dynamic response of components to pulses of broader length range, the additional capabilities of this machine can be readily employed.



B. DETERMINATION OF THE MECHANICAL PROPERTIES OF Ni-Cd ELECTRODES

1. Introduction and Model Analysis:

The mechanical properties of battery electrodes must be known before the engineering design of an impact-resistant cell can begin. At a minimum, static force data must be available to permit reasonable cell configuration design. A more refined design must be based on dynamic force data. An optimum cell design may indeed hinge on a complete knowledge of the mechanical properties of the fundamental structural elements of the battery. Of prime concern are the electrodes themselves. Their response to a dynamic force input must be determined from a knowledge of their stress-strain-time relation. A major effort under the current Jet Propulsion Laboratory contract was expended in gathering the experimental data needed to describe the mechanical properties of these elements. At the beginning of the experimental work, it was valuable to propose a model for the structural element we were investigating. We made some assumptions about our battery plate material, formulated a model and made some preliminary predictions from that model.

If we neglect the electrochemical differences between the positive and negative plates, we can think about



their basic structure alone. Both positive and negative plates begin as a sintered nickel structure called plaque. This structure is basically a 20 mesh screen of .007" diameter nickel wire to which particles of nickel powder have been sintered. Applied primary to one side of the screen, this sintered structure consists of particles of nickel powder sintered (fused) to each other and, of course, sintered to the screen. A first impression might be to observe the similarity of this plate to reinforced concrete. This comparison is not totally valid since the sintered matrix of our plates is a highly porous structure.

Although electrochemically very desirable, the porosity of these plates makes structural analysis most difficult! We have shown (2nd JPL QTR Report 1967) that depending on positive or negative, cycled or sterilized, the total porosity may range from 27% to 59%. It is apparent that we are dealing with a highly complex structure having a random distribution of various size openings. Some recent pictures obtained by scanning electron microscopy have illustrated the complexity and randomness of this matrix or at least the surface characteristics.

Fortunately, for the development of our structural model, these properties of the final plate are probably of



little consequence. We are assuming that only the original sinter structure (plaque, common to all plates) is of load carrying concern. In other words, we are assuming that it matters little from strength considerations that we fill this basic sintered porous structure to various degrees with active chemical material which has little reinforcement capability (especially in tension). Accepting this assumption, we need to be primarily concerned with only the description of the sinter matrix itself. It is this more highly porous structure that we wish to describe via a structural model. The model should allow us to calculate the actual load carrying area within this structure. Then, using the mechanical properties of nickel metal, we may calculate the strength and modulus associated with our model. We considered various cubic crystals as representative of the nickel matrix. The "atoms" (of radius "a") normally placed in the basic crystal, were considered to be hollow spheres; normal void space within the crystal was considered to be solid nickel metal. Three were considered because they offered models of a matrix having 74% (FCC), 68% (BCC) and 52% (simple) porosity. Most emphasis was placed on the face centered cubic model.

The value of these porous models lies in our ability to look at various cross sections and calculate the load



bearing areas that are available. This was done for the three principal planes of each cubic structure. The overlapping of values obtained from such divergent models (of porosity) indicates something about the random distribution. Our real structure has been shown to contain a random number of various size pores. Therefore, one would expect to see mean values between the extremes of any crystal. This seems strengthened by the fact that the least dense (100) plane of a 52% porous model has a modulus lower than the mean plane (100) of a 74% porous model. We may modify our predictions further by three considerations: a) the real sinter structure (plaque) is 78-80% porous and is thus higher than the 74% porosity of the FCC Model, b) the "necks" carrying load between pores in the structure are essentially areas of infinitesimal gage length. Thus, elongation is impossible and considerable tension load bearing area can be expected to be lost very readily, c) plate material has many small cracks initially due to manufacturing processes. These considerations would tend to lower the apparent modulus and suggest that $E = 3.5 \times 10^6$ as predicted by the FCC (111) analysis would be most valid. This should be true especially in tension. In compression, however, we might suspect an increase. The model approach can also be used to predict values of plate strength. Using handbook values for



nickel metal these calculations were made. The previous consideration of premature yielding of infinitesimal "necks" indicates that some of these calculated quantities are of little value. Once yielding occurs, the matrix will be ineffective and total load should be carried by the nickel wires in tension.

Stress-strain curves were drawn to present upper and lower bounds of the real behavior. A curve for screen alone is completely valid as the lower bound since it assumes no strength associated with a matrix. The curve for FCC (111) is somewhat of an intuitive selection as the possible upper bound (primarily for tension). We expected then, to see experimental tensile data generate a curve very close to the upper bound.

Experimental Program Outline

The data required in support of a model for plate material covers a rather broad experimental program. We are immediately interested in obtaining a stress-strain curve in both tension and compression that will enable us to examine this model more closely and check the assumptions that have been made. Ni-Cd electrodes must be fully characterized with respect to their mechanical properties. What is essentially involved is a determination of the applicable stress-strain relation. However, there are a



great many areas and tests required to fully explore, this relation and obtain the kind of information required to permit the analytical prediction of their ability to withstand a shock environment. The scope of the desired information is outlined below:

STRESS STRAIN BEHAVIOR:

- A. Tensile
- B. Compressive
 - 1) Static
 - a) Complete "Shape" of Curve
 - b) Modulus of Elasticity
 - (1) ΔE vs e -level
 - c) Yield Strength
 - d) Ultimate Strength
 - e) Poisson's Ratio
 - f) Elongation
 - 2) Dynamic
 - a) Strain-Rate Effects
 - (1) Yield Strength
 - (2) Ultimate Strength
 - b) Stress Amplification
 - d) $\sigma_y = f(\omega_n)$
- C. Bearing Strength (Edge Crushing)
- D. Thickness Compression of Pack
- E. Coef. of Friction at Various Pressures
- F. Buckling at Various Spacing & Pressures



2. Mechanical Properties in Tensile Loading

a) Constant Deformation Static Testing

The initial studies of the mechanical properties of Ni-Cd electrodes involved the static determination of "conventional" stress-strain curves in tension. "Conventional" here means the engineering σ - ϵ curve as established by constant deformation loading. All test samples consisted of 1/2" wide x 6" long strips containing exactly ten (10) longitudinal screen wires. An Instron commercial testing machine was used with a cross head speed of 0.1 in/min over a 4" initial gage length for a constant strain rate, $\dot{\epsilon}$, equal to 0.0004 sec^{-1} . With cross head speed related to chart speed, the x-axis as deformation was established and with load cell output on the y-axis we received initial data in the form of a direct load vs deformation plot which was then easily converted to a stress-strain curve. For positive plate, negative plate, positive plaque and negative plaque the stress is simply the load divided by the actual measured area of the specimen. However, for screen specimens, an arbitrary apparent area of 0.013 in^2 was taken to permit strength comparisons with the final plate material. What we sought from this type of testing was the comparative strengths of the



screen alone, the reinforcement associated with the sintered nickel matrix and finally the actual strengths and differences in strength of positive and negative plate. Thus, the rough determination of modulus of elasticity, the engineering yield strength (0.2% ϵ), the ultimate strength, elongation (% in 4" G.L.) and most importantly, the complete shape of the stress-strain curve into the plastic region were of immediate concern. Several specimens were tested of each of the five "materials" of interest.

Reproducibility is excellent in this type of loading and the average results for each material are shown tabulated below with respect to important values describing the mechanical behavior.

TABLE III B-1

	<u>ANNEALED SCREEN</u>	<u>NEGATIVE PLAQUE (905)</u>	<u>POSITIVE PLAQUE (810)</u>	<u>* NEGATIVE PLATE (170)</u>	<u>* POSITIVE PLATE (176)</u>
E (PSI)	0.18×10^6	0.23×10^6	0.47×10^6	0.41×10^6	0.58×10^6
ρ_{yield} (LBS)	2.9	5.1	11.1	10.0	11.05
σ_{yield} (PSI)	225	390	855	710	850
ρ_{ult} (LBS)	16.8	17.3	18.9	19.1	18.1
σ_{ult} (PSI)	1290	1330	1450	1360	1390
Elongation (%)	20.7	22.8	17.7	18.5	20.5

* Dry, as manufactured electrodes

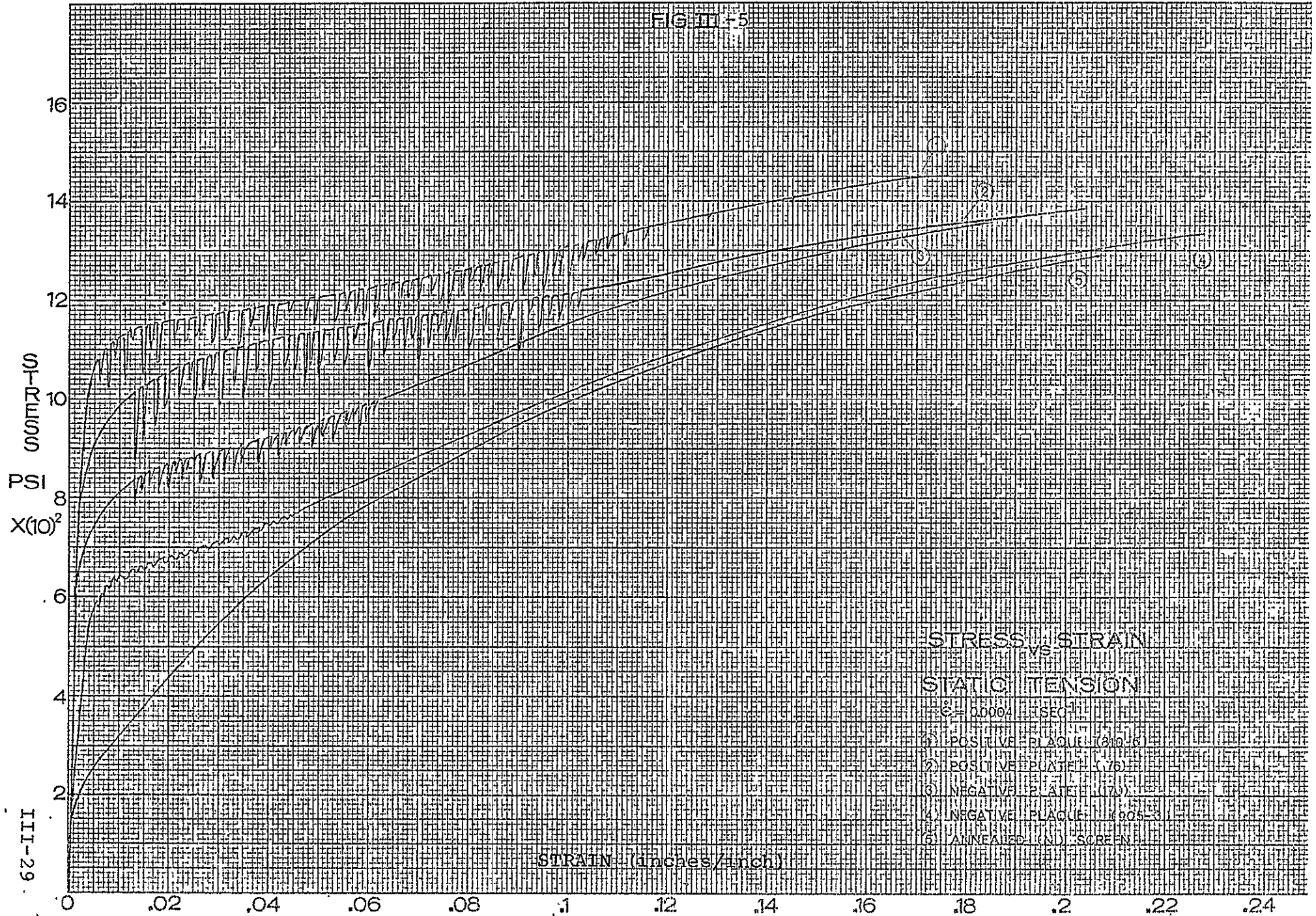


In order to summarize the results of this study more clearly, Figure III-5 has been prepared to show the average stress-strain curve obtained for each material studied. The curve for screen alone essentially lays the foundation of comparison in this study. We can observe the progression of increased strength from the basic screen grid through negative plaque, negative plate, positive plate and finally positive plaque. It is desirable to compare these various materials with regard to three important quantities:

Modulus of Elasticity -- The values for each material were tabulated as an indication of relative strength. They were determined from approximate slopes of the initial portion of the σ - ϵ curves. They are valid as a confirmation of relative strength but tend to be low in absolute value. Section B.2.b of this report describes in greater detail the results obtained from a more precise method of determining this quantity.

Yield Strength -- The prime consideration in discussing yield strength is the degree of basic grid reinforcement that is attributable to the addition of the matrix. As first applied to the basic grid, this matrix of sintered Ni powder forms plaque. The first important result of this study was the significant difference in strength between positive and negative

FIG. II-5





plaque. Positive plaque has a much higher yield strength than negative plaque. Although both materials are manufactured from the same slurry formation and have the same thickness, the positive has a higher area density than the negative. The positive plaque has less total porosity than the negative plaque and in addition, the sintering temperature is higher for positive plaque than for negative plaque. These facts explain very readily the reason for the increased strength of positive plaque.

The next significant effect shown is the reduction in strength that occurs when positive plaque is electrochemically formed into positive plate. Porosity alone is certainly no indication of this effect for it is not uncommon to reduce the total porosity. The factor that is important is the corrosion of the sinter matrix which occurs during the process. It is true that positive active material is deposited within the pores and results in the lowered porosity, but it appears that this material in hydroxide compound form adds little strength to the material. The overriding effect, then, is the significant weakening of the initially strong positive plaque matrix by the extensive



corrosion associated with impregnation.

Equally significant is the reverse strength effect we have observed associated with the manufacture of negative plate. Negative plate material has a higher yield strength than the initial negative plaque. The total porosity is decreased during this process but again, it is the manner in which this change occurs that is important.

Firstly, however, the amount of sintered matrix corrosion is less for the negative plate than for the positive plate. Secondly, the negative active material partially as a metal, has more strength than the positive compounds. It is, therefore, apparent that the lower amount of corrosion combined with the overriding factor of a stronger active material produce the net strengthening of negative plate. It is shown as a final result that positive plate material has a higher yield strength than negative plate material. This is obvious at the 0.2% ϵ point, by the higher modulus of elasticity, and by the larger drop of load as the matrix cracks. Not only is the positive plate matrix stronger, but it continues to help support load further into the high strain region. It seems then, that even after considering the differences in



corrosion and impregnation phenomenon, the positive Ni plate is stronger than the negative Cd plate primarily because there is such a great difference in their respective parent structures -- the positive and negative sinter material -- plaque.

Ultimate Strength -- Although the basic progressive ineffectiveness of the matrix has been shown by the reversion of plaque/plate strength to that of screen alone, it is first important to note that the ultimate strength of all four matrix materials remains slightly higher than that of screen alone. This may be accounted for by assuming that some of the nickel powder has been effectively sintered to the screen wires or in the immediate vicinity and permits this small portion of the matrix to provide continued reinforcement throughout elongation to fracture. If this is indeed the mechanism acting, we can then also explain the relative values of ultimate strength from the corrosion consideration discussed earlier. Firstly, it is reasonable to expect the higher density, higher temperature sinter of the positive plaque to exhibit the highest ultimate stress which indeed is the case. Secondly, we note that despite a stronger matrix at yield, the positive plate has a lower ultimate strength (absolute load) than the



negative. Here the greater corrosion of the positive wires and proximate matrix would tend to weaken and lower the ultimate strength below that of the negative plate.

b) Constant Load Static Testing

Although the previous study is an excellent starting point for an investigation of a material's mechanical behavior and provides the usual engineering understanding, it may not provide the optimum information for our purposes. Thus, we employed a constant load test frame rather than the commercial constant deformation type used in the preliminary studies. This arrangement applied load to the specimen by means of a water-fill technique. A container attached to the specimen served as a dead weight accumulator as it was filled with water at a constant rate. Additionally, unloading at any point along the stress-strain curve is easily accomplished at a fairly uniform rate by draining water from the accumulator. Load is measured by a strain gage proof-ring and the signal plotted directly on the vertical axis of an x-y recorder. Strain measurements are taken more accurately by a strain gage extensometer of 1" gage length mounted on the specimen. This output is plotted directly on the horizontal axis of



the x-y recorder. This system provided a very deliberate manner of loading to closely examine the region of yielding. Especially because of our ability to unload very precisely, we were also able to examine the modulus of elasticity of the material and its relation to strain level. Again, the average results for each material are shown in Table III B-2 with respect to the important values describing its mechanical behavior.

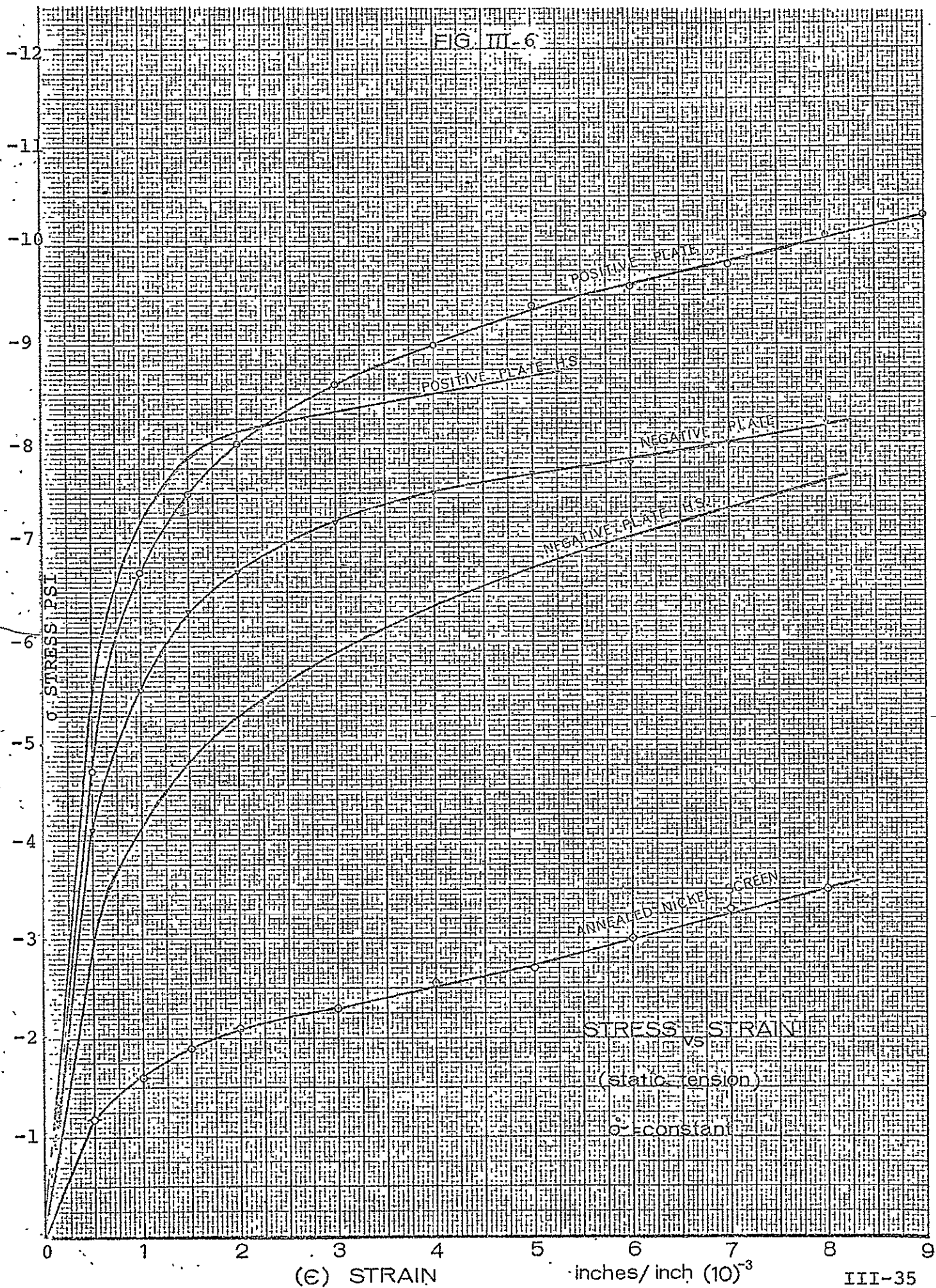
TABLE III B-2

	<u>ANNEALED SCREEN</u>	<u>* NEGATIVE PLATE</u>	<u>* POSITIVE PLATE</u>
E (PSI)	$0.8 \pm 0.2 \times 10^6$	$1.0 \pm 0.1 \times 10^6$	$1.2 \pm 0.2 \times 10^6$
ρ_{yield} (LBS)	3.1	9.9	11.2
σ_{yield} (PSI)	240	705	860
$\rho_{\text{ult.}}$ (LBS)	17.0	19.8	19.1
$\sigma_{\text{ult.}}$ (PSI)	1310	1420	1470

* Dry, as manufactured electrodes

Figure III-6 has been prepared to permit comparison of the average stress-strain curve associated with each material studied. It is shown once again that the screen alone forms the foundation and the

FIG. III-6





increased strength due to the matrix is very obvious in this region of low strain ($<1\%$). The tabulated values of strength agree quite well with those obtained under the constant deformation type of loading. However, this description of strength was obtained in a different manner and it is best to consider each value separately.

Modulus of Elasticity

As the matrix becomes ineffective, (while cracks form as strain level progresses), we would expect to see a progressive decrease in the initial modulus value toward the lower value of screen alone. As indicated, the constant load technique allows us to unload and reload at any strain level we chose to determine the elastic modulus at that strain level. In practice, we were able to unload and reload at $400 \mu \text{ in/in} < \epsilon < 6000 \mu \text{ in/in}$. Within this region, the experimental scatter is equal to the diminishing function which may exist. Some values were obtained near the $200 \mu \epsilon$ level, and they were higher than the average reported. The values that have been given in Table III B-2 are the average values obtained from the unload-reload determinations within $400 \mu \text{ in/in} < \epsilon < 6000 \mu \text{ in/in}$ for each material. The tolerance is that of



experimental error combined with any diminishing function that may exist within this region of strain. We have, of course, plotted E vs. ϵ for each material and can only say that three regions of interest seem to exist: (a) E may be diminished rapidly in the region $0 < \epsilon < 200\mu$ in/in; (b) for the region 200μ in/in $< \epsilon < 2000\mu$ in/in there may exist a very small diminishing of E ; (c) beyond $\epsilon = 2000\mu$ in/in there seems to be no function of E vs ϵ level. In other words we feel that a plastic deformation component exists at very low strain levels ($< 200\mu$ in/in) and that yielding is complete @ $2000\mu\epsilon$. This is confirmed by the fact that a determination of modulus of elasticity by measuring the slope of the stress strain curves will give a lower value than that obtained by the more accurate unload-reload technique. The values indicated in Table III B-2, therefore, remain the best measure of true elastic response in tensile loading. Aside from permitting future calculations of dynamic response, they are important measurements for several reasons: (a) the value obtained for screen alone agrees well with that indicated by the model; (b) at least for $\epsilon < 6000\mu$ in/in the modulus of plate material remains higher than that of screen alone; (c) the strength (stiffness)



of the positive matrix is greater than that of the negative matrix.

Yield Strength

As indicated, this type of precise loading was meant to more accurately examine the region of yielding $\epsilon < 1\%$. Figure III-6 illustrates this region in great detail. It is important only to note that this value of yield strength @ 0.2% ϵ agrees very well with the initial determination given in Table III B-1. Thus, this series of constant load tests fully supports the initial relations between positive and negative plates as extensively discussed in the previous section.

Ultimate Strength

Although this type of testing was meant to closely examine only the low strain region near yielding, most specimens were taken to tensile fracture. Load data was available up to the point of failure and it is important only to note that the ultimate strengths thus reported are essentially in agreement with those first determined and reported in Table III B-1. Thus, this series of constant load tests essentially supports the discussion of ultimate strength contained in the earlier section.



c) The Effects of Heat Sterilization and Electrolyte Saturation

All tabulated values and stress-strain curves shown in the preceding sections were obtained from as-manufactured, dry, unsterilized plate material. We were interested, therefore, in the possible changes in mechanical properties which might result from heat sterilization and electrolyte saturation of this same plate material. Tensile samples were thus prepared by full heat sterilization (135°C for 64 hours) in the saturated condition (30% KOH). These specimens were then tested in the constant loading apparatus described in Section B.2.b. The results are tabulated below and should be compared to those in Table III B-2.

TABLE III B-3

SATURATED AND STERILIZED ELECTRODES

	<u>Negative Plate</u>	<u>Positive Plate</u>
E (psi)	$0.71 \pm 0.10 \times 10^6$	$1.35 \pm 0.25 \times 10^6$
ρ_{yield} (lb)	8.2	10.9
σ_{yield} (psi)	587	835
ρ_{ult} (lb)	19.2	22.0
σ_{ult} (psi)	1375	1690

The complete stress-strain curves have been included in



Figure III-6 in order to permit the comparison of as-manufactured vs saturated and heat-sterilized plates.

It appears that the mechanical properties of the positive plate material are not changed as greatly as those of the negative plate. The positive results indicate very small changes in the modulus of elasticity and yield strength. However, the ultimate strength has been found to be significantly higher.

The behavior of negative plate is changed a great deal by this saturated sterilization. The modulus of elasticity and yield strength of the negative has been significantly lowered along with a slight reduction in ultimate strength. The weaker of the two electrodes in our Ni-Cd system seems to be further weakened by the saturated sterilization treatment that is required.

d) Dynamic Tensile Testing

Considering the goal of the current effort to develop knowledge which will enable the design of impact-resistant Ni-Cd cells, it should be stated that the actual dynamic tests are of prime importance. In retrospect to what has been written, it should be stated that the reason for performing static testing is to establish a foundation of strength properties. What is necessary for a thorough understanding



is a knowledge of the actual transfer from static to dynamic conditions. The static testing is valuable since it can provide us with a complete picture of stress vs strain which is not easily obtained under dynamic loading. In fact, it will be shown that only the yield point and ultimate load can be obtained under dynamic conditions and that the static results are of great value in establishing the complete mechanical behavior.

Ultimate Strength:

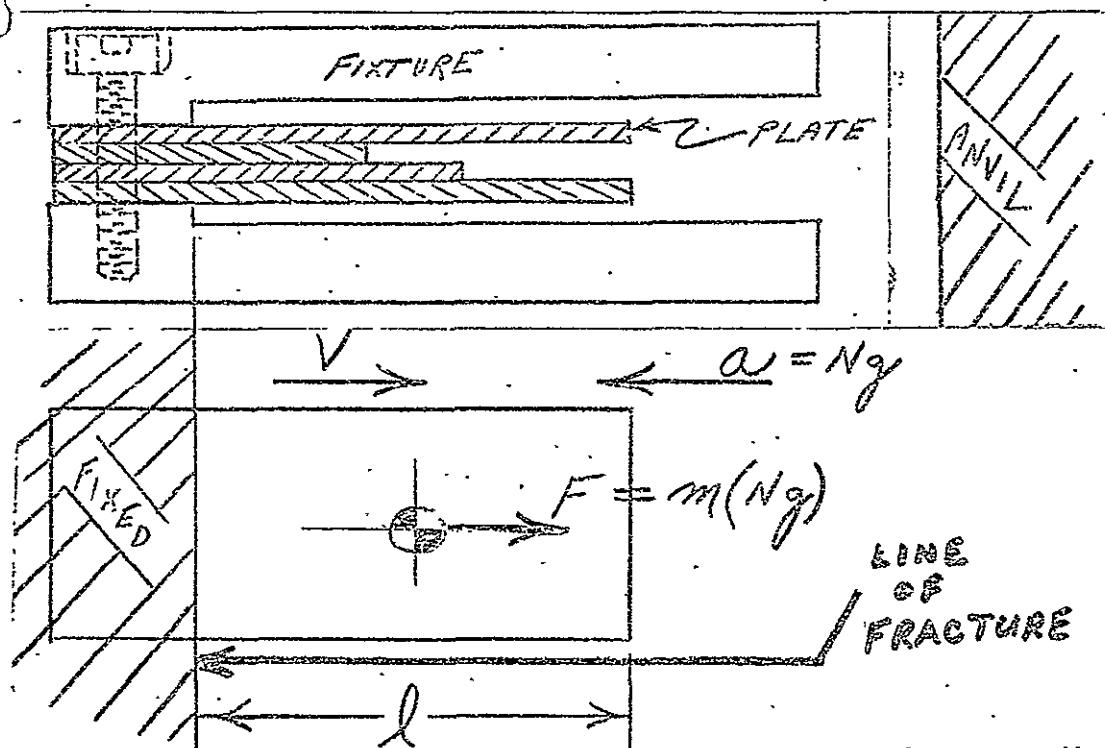
From the static values we have obtained and a knowledge of the plate's physical properties, we might write --

$$\left(\frac{\text{Ultimate Force}}{\text{Unit Width}} \right) = \left[\frac{(\text{Length}) \left(\frac{\text{Unit}}{\text{Width}} \right) \left(\frac{\text{Area}}{\text{Density}} \right)}{g} \right] (\text{N g})$$

and would find that --

$$l \geq 3.3 \text{ inch} \Rightarrow \text{Failure}$$

This mass, inertia-loading defines the method of testing we have used to investigate dynamic properties. The essential concept is shown diagrammatically below.



All specimens of positive and negative plate material were constant width (2" or 40 wires) and cut to various lengths (3-1/2" - 6-1/4"). Typically, four positive plates and four negative plates, each of a different length, would be placed in the fixture. Through suitable shims, friction duplication techniques (support data of frictional coefficients have been generated) and clamping arrangements, we could consider one end fixed. Then, knowing the effective free length, l , the mass density, ρ , and the mean g-level at impact, Ng , we could calculate for each plate the dynamic tensile inertia force experienced. Therefore, lengths were varied above and below the predetermined static failure length and the value was



bracketed. Values of ultimate strength are tabulated below for comparison.

TABLE III B-4

ULTIMATE TENSILE STRENGTH (2" WIDTH)
(As-Manufactured Electrodes)

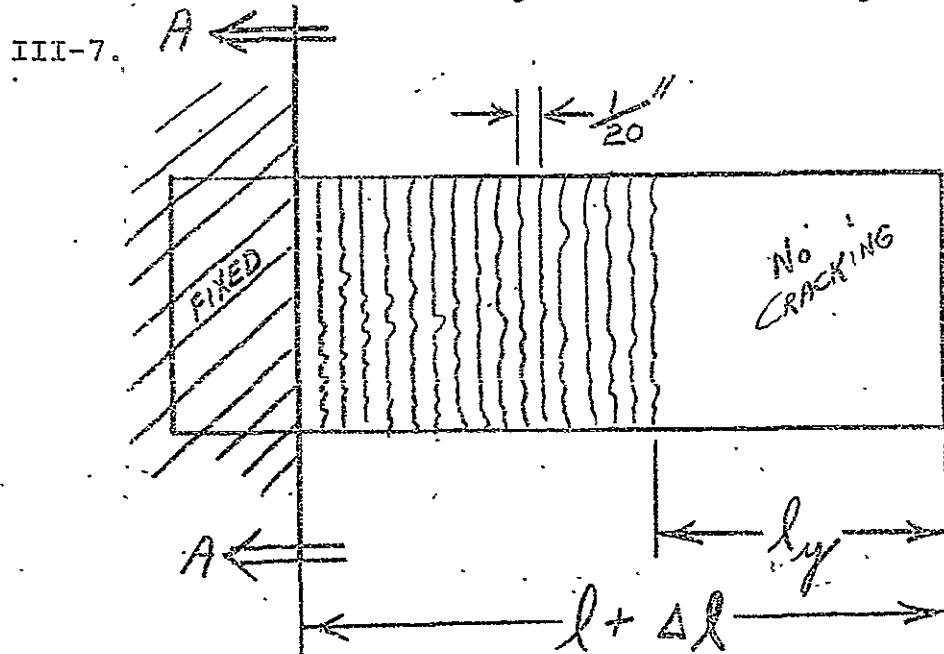
	ϵ Static <u>ϵ = Constant</u>	σ Static <u>σ = Constant</u>	Dynamic <u>7/8" Tool</u>	Dynamic <u>1" Tool</u>
Positive Plate	72#	76#	80#	107#
Negative Plate	76#	79#	90#	115#

What is apparent from this study is that the plate material can support greater loads under dynamic conditions and we indeed may use static strength determinations as a foundation or lower bound for design. In other words, the material is ϵ sensitive and under dynamic loading may support perhaps 120#. The reason that the dynamic tests as reported above are separated is due to the different pulse lengths involved. The calculated values shown are magnified to some degree by the dynamic loading. It is felt that we are in the area where the magnification is not constant and is dependent upon the natural and applied frequencies involved. This aspect is discussed in Section C.2.b of this report.



Yield Strength

In addition to bracketing the value of ultimate strength under dynamic loading, we are also able to determine the material's dynamic yield strength, as indicated by the onset of matrix cracking. For example, after impact loading a plate has the appearance shown in the following sketch and in Figure

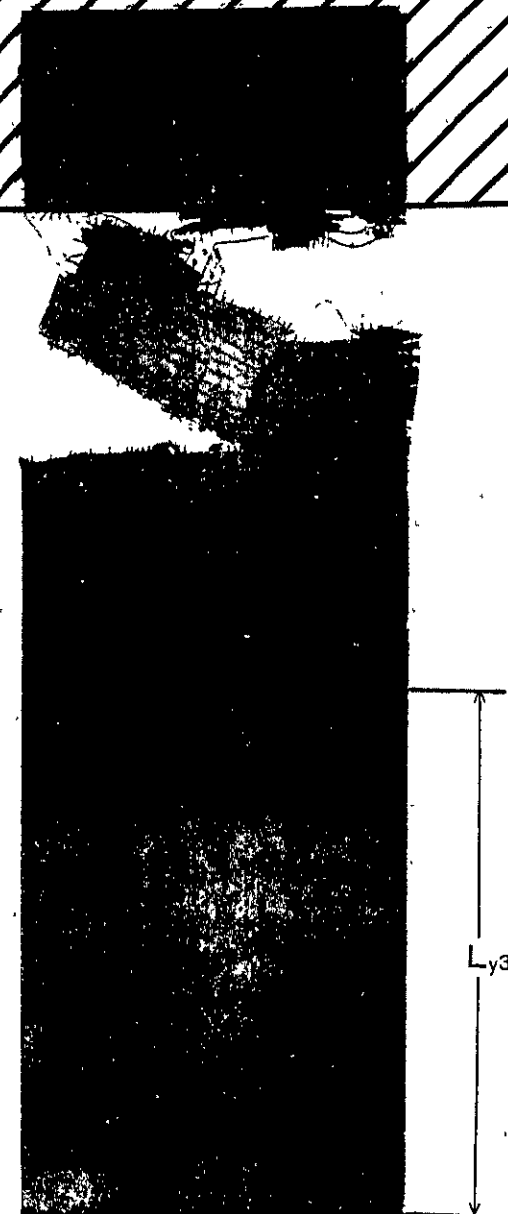
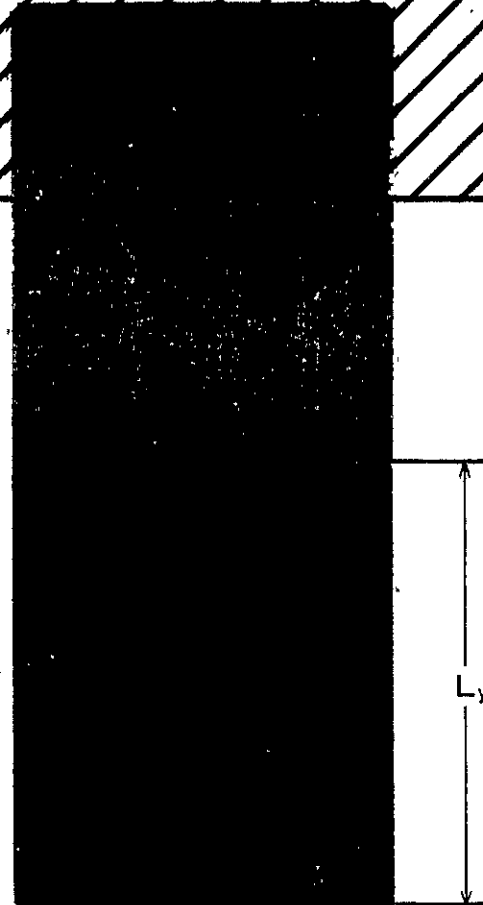
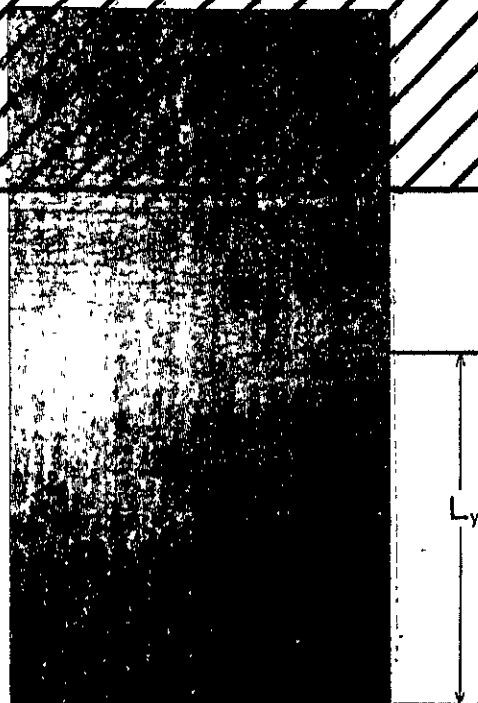


As stated previously, the value of effective length, l , will determine whether or not there is sufficient mass experiencing the test acceleration to produce a force sufficient to fracture the plate at Section A-A. There also exists for each plate a length, l_y , from the free end over which no matrix cracking occurs. Beyond l_y the matrix will have cracks uniformly spaced, each $1/20$ inch (20 mesh screen). If we measure l_y for each plate, then, knowing the input

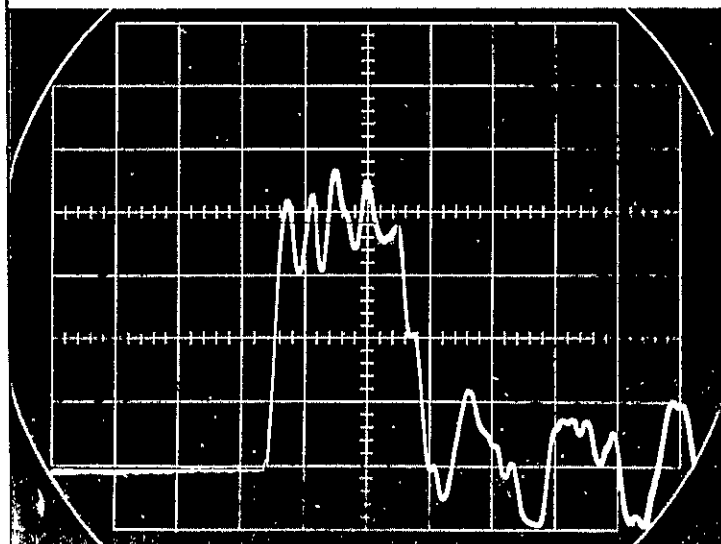
DYNAMIC TENSILE LOADING



VELOCITY
114.5 FT/SEC



$$L_{y1} < L_{y2} < L_{y3}$$



1.05 msec

3850 g's mean

Figure III - 7



acceleration and the physical properties of the plate, we can calculate the dynamic force created by this mass which was responsible for yielding the plate or cracking the matrix. This dynamic yield strength has been calculated for each plate tested. We have plotted the yield force, ρ_y , against the effective plate length, l , to demonstrate the distinct linear function that exists because of relative frequency - magnification considerations. The effect is amplified by our failure to consider the progressive nature of yielding. The range of yield strength values are given in Table III B-5.

TABLE III B-5
YIELD STRENGTH (2" WIDTH) (AS-MANUFACTURED ELECTRODES)

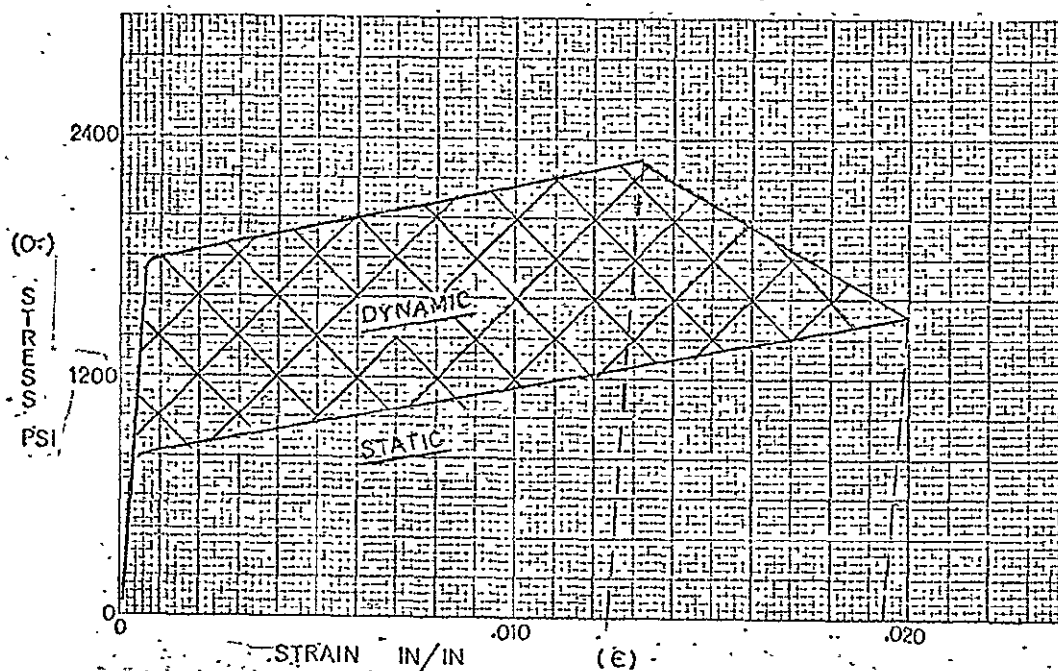
	$\sigma = \text{Constant}$ $\rho_y @ 0.2\% \epsilon$	$\sigma = \text{Constant}$ $\rho_y @ 0.2\%$	Dynamic 7/8" Tool $\rho_y @ \text{Crack}$	Dynamic 1" Tool $\rho_y @ \text{Crack}$
Positive Plate	44#	45#	60-72#	42-96#
Negative Plate	40#	40#	60-70#	48-78#

Again, it is apparent that yield strength as well as ultimate strength is elevated by the higher strain rate associated with dynamic loading. The



variation with pulse length (7/8" vs 1" tool), and effective plate length has been demonstrated.

Rather complete static stress-strain curves have been established and the effect of strain-rate sensitivity has been shown:



Dynamic strength behavior is elevated above that of the well-established static lower bound. However, the degree of this elevation (shaded area above) depends on a more well-defined relation between element response and the applied forcing function.

e). Poisson's Ratio

Extensive development of strain gage techniques was carried out under the present contract. The supporting data can be found in the 9th QTR JPL Report.

The prime reason for development was the measurement



of dynamic strain during impact conditions as discussed in Section C.2.b. These techniques also offer the most appropriate method that can be used for a determination of Poisson's Ratio. The static determination employed an axial strain gage and an identical transverse strain gage of the thin paper foil type installed with Eastman 910 on each of three $\frac{1}{2}$ " wide dry negative plate specimens. Simultaneous strain readings were taken from each gage and a commercial axial extensometer. Essentially, we compared the slope ratio of the transverse to the axial gage output within the region of linear agreement between the axial gage and the axial extensometer. The constant loading as described in Section B.2.b was used for these tests. The third specimen was subjected to one unload-reload cycle to show the elastic behavior more clearly.

The results are presented in Table III B-9. We have indicated the modulus of elasticity, E , as determined at various strain, ϵ , levels by unload-reload cycles. The values obtained are in good agreement with those previously determined by this type of loading. We have also given the calculated ratio of strain gage output to that of the

TABLE III-9

Negative Plate Specimen $\frac{1}{2}$ " Wide	E @ ϵ in/in (PSI)	($\frac{\epsilon \text{ axial gage}}{\epsilon \text{ axial extensometer}}$) ϵ -limits μ in/in	($\frac{\epsilon \text{ transverse gage}}{\epsilon \text{ axial extensometer}}$) ϵ -limits μ in/in	Poisson's Ratio ν
5-28	1.22×10^6 @ .0014 1.00×10^6 @ .0038	0.702 (50-300)	0.116 (50-350)	0.164
6-16	1.30×10^6 @ .0044	0.400 (100-600)	0.088 (100-700)	0.218
Initial Loading	-----	0.364 (80-600)	0.068 (20->625)	0.186
6-27 Reload @ 0.0012 in/in	1.01×10^6 average @ .0012	0.733 (10->300)	0.124 (40->300)	0.169



extensometer. Next reported, are the values of Poisson's Ratio, which were obtained from each test. The average value is $\nu = 0.184$. However, more weight must be given to the values obtained at the higher ($\frac{\epsilon_{\text{axial gage}}}{\epsilon_{\text{axial exten.}}}$) ratio since this indicates a lower degree of matrix reinforcement. On this basis, the best value of Poisson's Ratio is approximately 0.17 and should be used for subsequent calculations. This value is of course much lower than the 0.28 normally observed with pure nickel and indicates a large degree of anisotropy. We had previously observed the differences in tensile and compressive strength, and this predicted anisotropic behavior has been confirmed under this uniaxial loading study.

3. Mechanical Properties in Compression Loading

a) Introduction

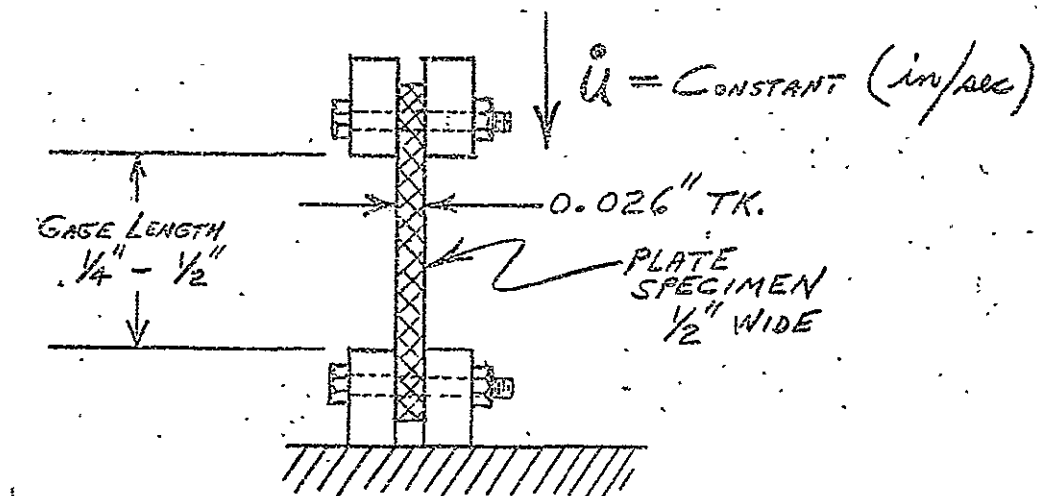
The experimental program for evaluating the mechanical strength of Ni-Cd electrodes may be broadly separated into three (3) distinct methods of loading: tensile, compression, bearing. The behavior under tensile loading has been discussed in Section B.2. The studies of bearing strength are of prime design interest and will be discussed in Section B.4. This section describes the manner of loading and the strengths that have been measured in a com-



pression-buckling mode of failure. This area of investigation is similar to the tensile work in that it has primarily increased our understanding of our plates' behavior under load. In contrast, the bearing strength studies will be more clearly related to actual cell failure mechanisms. In other words, edge crushing of the electrode pack could be a prime cell failure mode under impact loading.

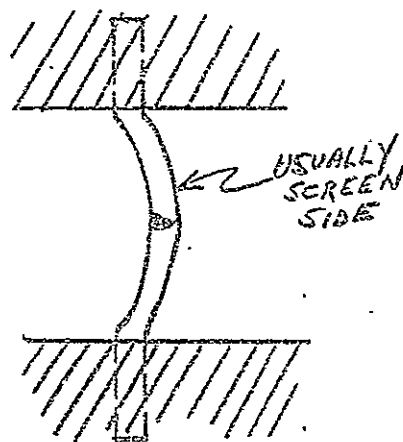
b) Constant Deformation Static Testing

The static compression loading was accomplished by the constant deformation method (commercial testing machine) as fully described in the tensile report. The manner of specimen loading is shown below:

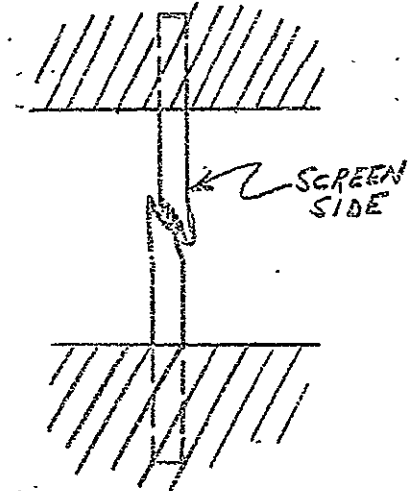




Two prime failure modes were observed, during the course of our investigation:



BUCKLING



COMPRESSION - SHEAR

Since the loading system did not provide lateral support, buckling would occur whenever the effective column length was too large for the material strength available. If the material has sufficient strength, buckling is resisted and the higher compressional ultimate load is exhibited.

In the first series of tests, as-manufactured positive and negative electrodes 1/2" wide were loaded over a 1/2" initial gage length. In all cases, the buckling mode of failure was exhibited.

The second series of tests were performed on 1/2" wide specimens of positive and negative plaque over a 1/4" gage length. Again, the universal failure



mode was in buckling.

The third series of tests involved as-manufactured positive and negative plates $1/2$ " wide loaded over a $1/4$ " gage length. This was a sufficiently short gage length to permit the specimens to resist buckling and primarily exhibit compressional shear failure.

Reproduceability of stress-strain information for various specimens of the same material is poor when obtained by this manner of loading. Figure III-8 presents, therefore, average stress-strain curves for the three series of tests. The positive (both plate and plaque) is shown to be stronger than the negative in all cases. The very large increase of plate strength over that of plaque is shown. The absolute stresses shown in Figure III-8 are significant. Regardless of mode of buckling-compressional failure, we observe ultimate stresses of 3000-6000 psi as compared to 1400 psi tensile ultimate stress. What is needed for true compressional evaluation, however, is a measure of the uniform strain with a provision for positive lateral restraint.

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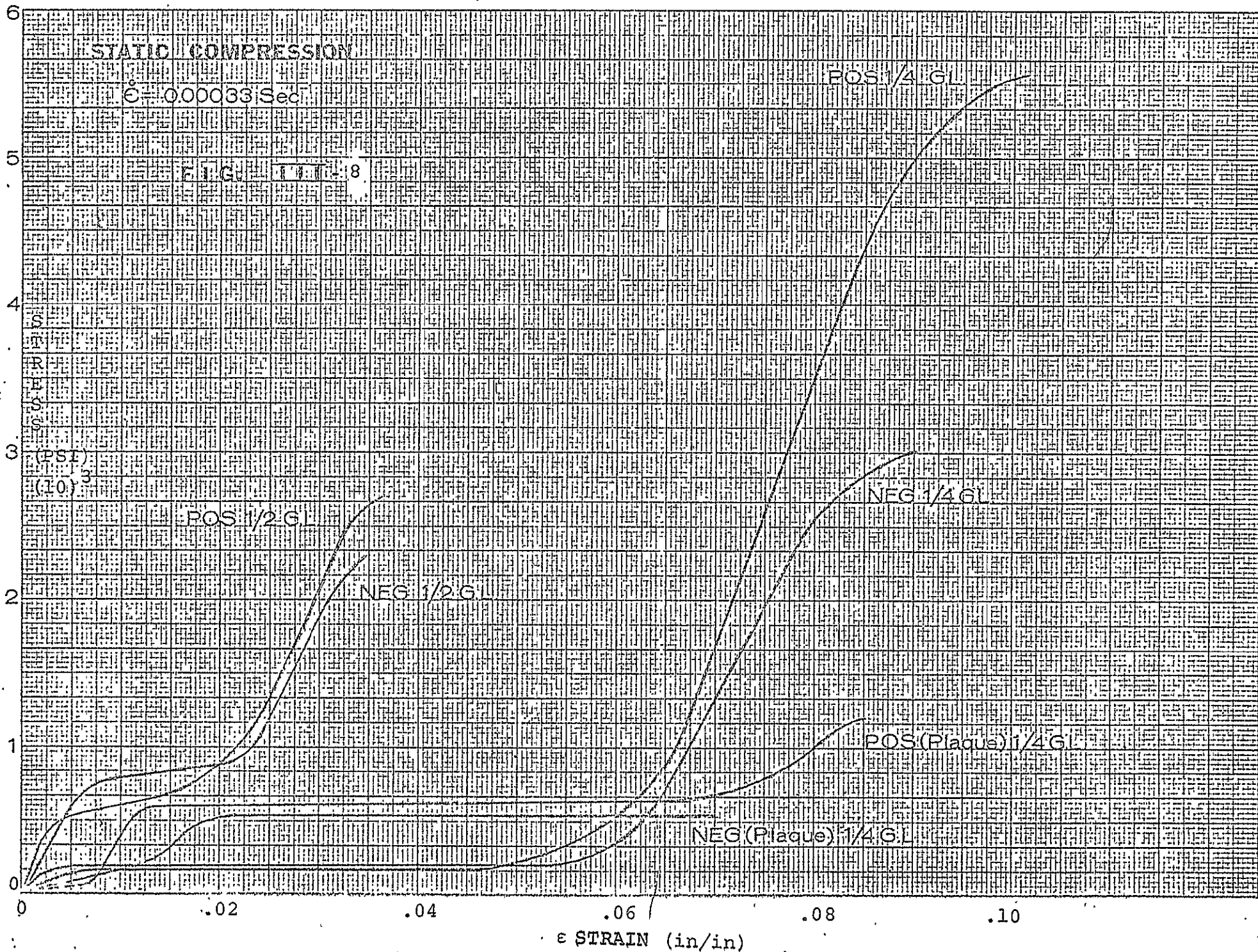


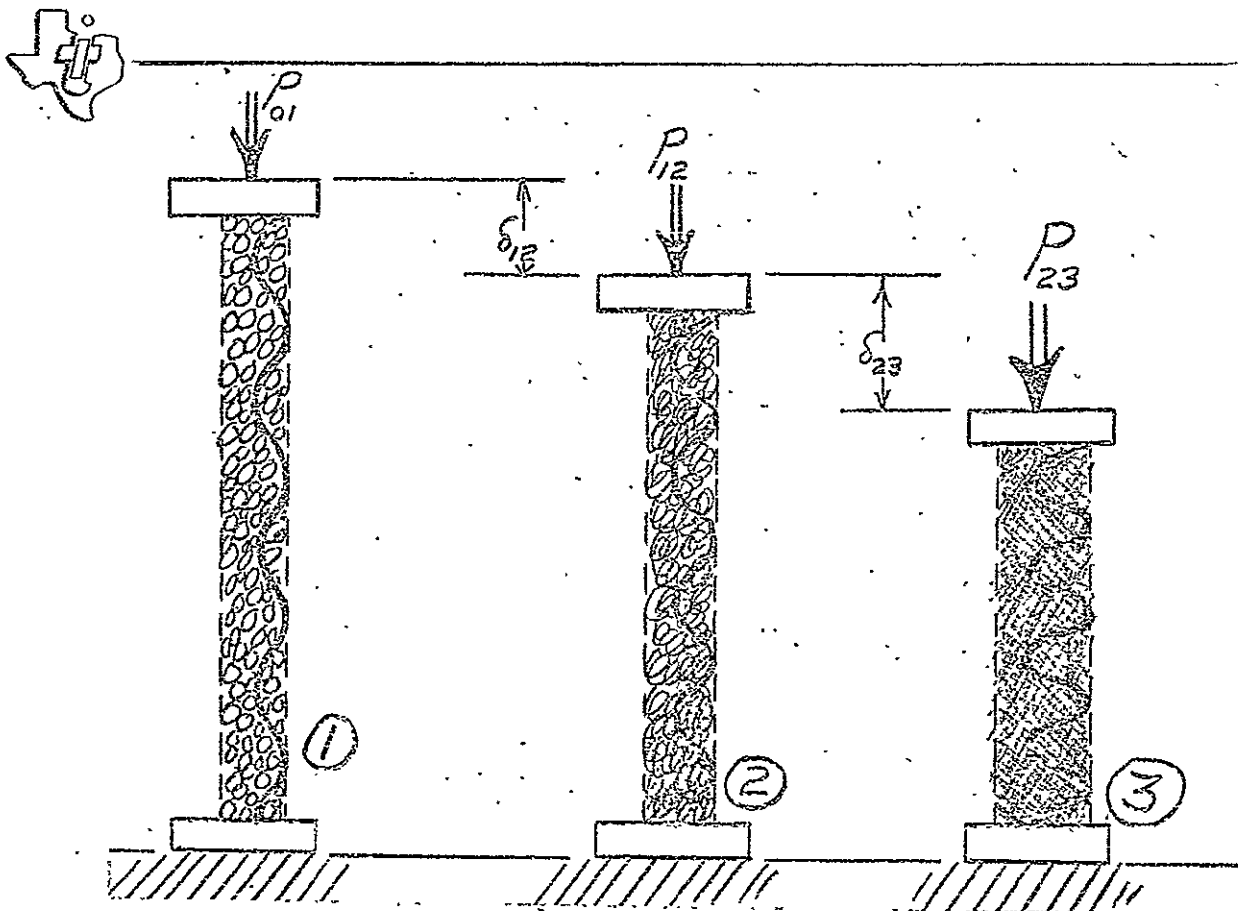


Table III B-6 compares the experimental values obtained with some fundamental buckling calculations. E_1 represents the initial apparent modulus of elasticity if significantly exhibited before yield. Following the plastic elongation region, the final "apparent modulus of elasticity" displayed during ultimate loading is represented by E_2 . The actual failure load P , is next given but distinguished by the associated mode of failure. Next we have shown a predicted buckling load as calculated by a modified Euler equation which serves merely as a very inaccurate lower limit for comparison purposes.

The absolute values of E_2 noted in this type of testing are lower than those observed during constant deformation tensile testing. It also appears from this study, that a rather large amount of plastic deformation is necessary before the matrix is able to support the load. Such factors suggest that the initial bends in the screen play a very significant role.

TABLE III B-6

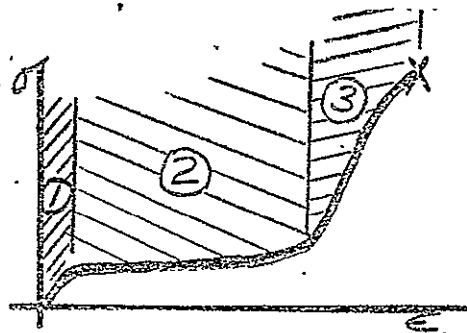
		E_1 $\times 10^6$ psi	E_2 $\times 10^6$ psi	Static (P) Critical Buckling #	Static (P) Ultimate Compression #	Calculated (P) Critical Buckling $2 \left(\frac{\pi^2 E_2 I}{12} \right)$	Dynamic (P) Ultimate Compression #
Positive Plate	1/2"	0.24	→ 0.12	35 ↑	----	8	/
Negative Plate	G.L.	0.12	↔ 0.12	32	----	8	
Positive Plaque	1/4"	0.14	→ 0.05	15 ↑	----	12	/
Negative Plaque	G.L.	----	0.10	8	----	24	
Positive Plate	1/4"	0.14	0.18	---	70 ↑	42	> 34
Negative Plate	G.L.	0.16	0.16	---	42	36	34



Porous structure uncrushed. Supporting initial load as in tension.

Porous structure has reached yield, Pores collapsing, wire bending.

Sufficient deformation & densification of collapsed pore structure to permit further load support.



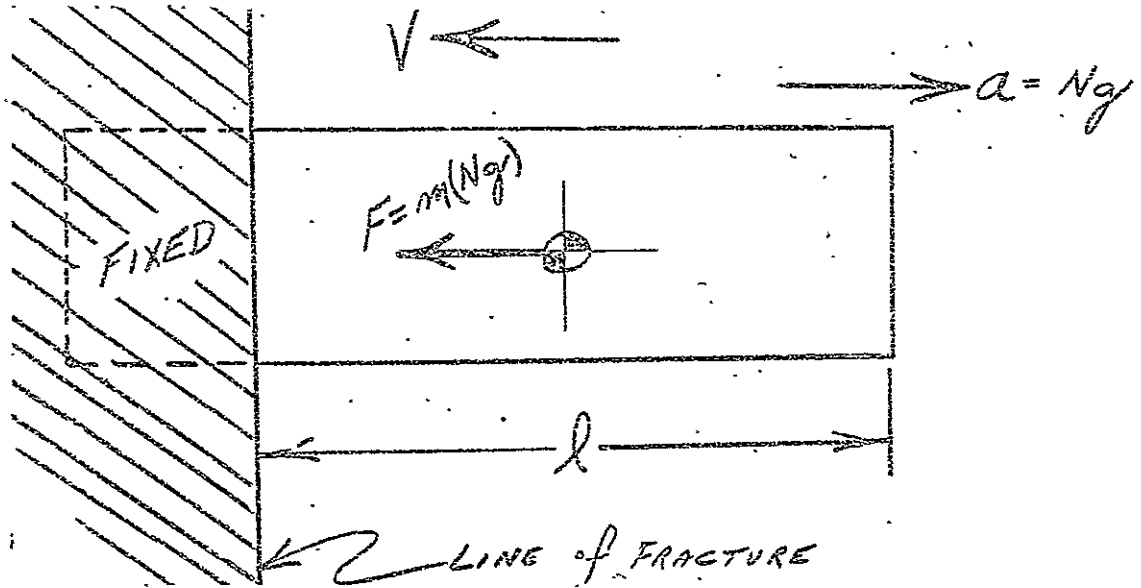
This understanding or explanation is supported by several experimental results. Yield strengths on the order of 100-600 psi have been observed in compression which are lower than those (225-850 psi) experienced under tensile loading. The extensive



plastic elongation region is then fairly well established. The modulus E_2 observed when load is again supported is \leq the modulus of screen alone under tensile loading. This indicates that the contribution of screen under compression is reduced and that indeed from a modulus standpoint, the combination of corrugated screen and compressed sinter are scarcely equal to the screen alone in tension. The ultimate load carrying ability in tension was seen to rely primarily on screen alone since the matrix became ineffective or lost due to elongation. In contrast, however, compression appears to utilize the matrix as the major load bearing element up to the point of fracture.

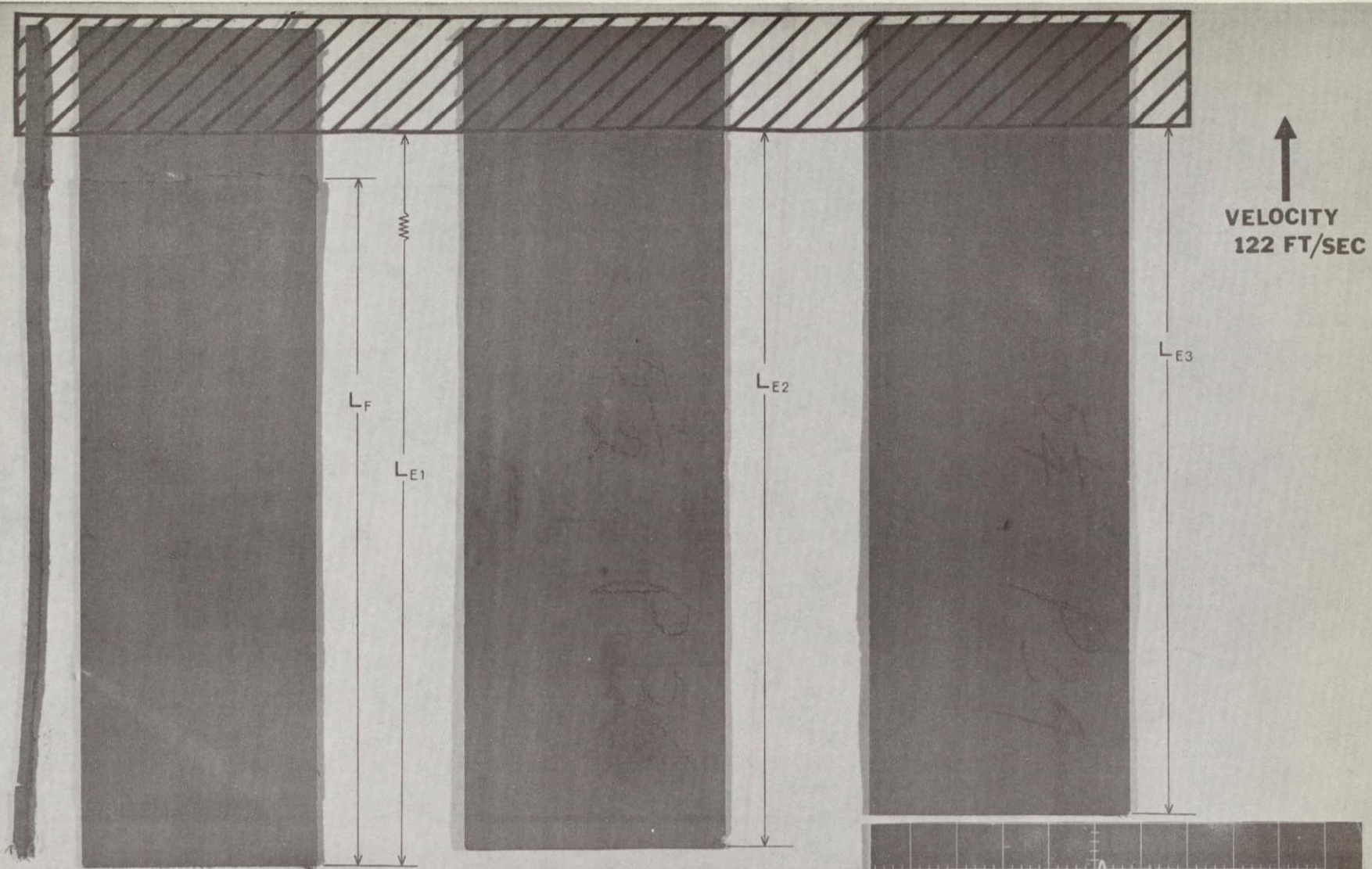
c) Dynamic Compression Loading

The dynamic test method used to evaluate Ni-Cd plates has been completely described in the tensile loading Section B.2.d. The method used for compression studies is identical except for a reversed mounting procedure which simply allows the plates to experience decelerating forces in the opposite direction.



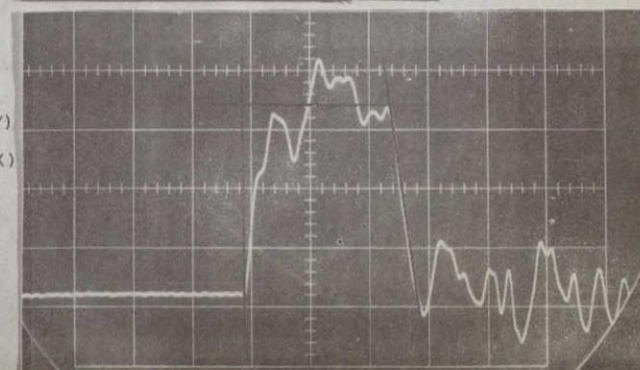
As in tension testing, the compressive specimens were 2" wide and cut to provide various effective lengths, l , for each impact. Through the use of suitable clamping and friction elimination techniques, each plate was essentially free to exhibit its response to the input force. A buckling mode of failure was not permitted, and thus we observed only compressional fracture as shown in Figure III-9. If the effective length, l , was sufficiently large, failure would occur at the associated loading. A dynamic yield point is not observed in the compression loading of the matrix as was seen in tensile loading. Thus, we have merely a determination of the ultimate compression load.

Positive Plate could not be failed in a completely dynamic compressional mode in our testing. We have



$$L_{E1} > L_{E2} > L_{E3}$$

1000 G'S/CM (Y-Y)
1.25 MSEC/CM (X-X)



DYNAMIC COMPRESSIVE LOADING

Figure III - 9



observed plates of the maximum effective length permissible ($l = 5-7/16$ ") experiencing impact levels of .3700 g with no apparent damage in pure dynamic compression. Thus, we may only say that the $P_{ult} > 134\#$ or $\sigma_{ult} > 2600$ psi. Indications are then, that the positive is stronger than the negative which is in agreement with the very obvious result of static testing.

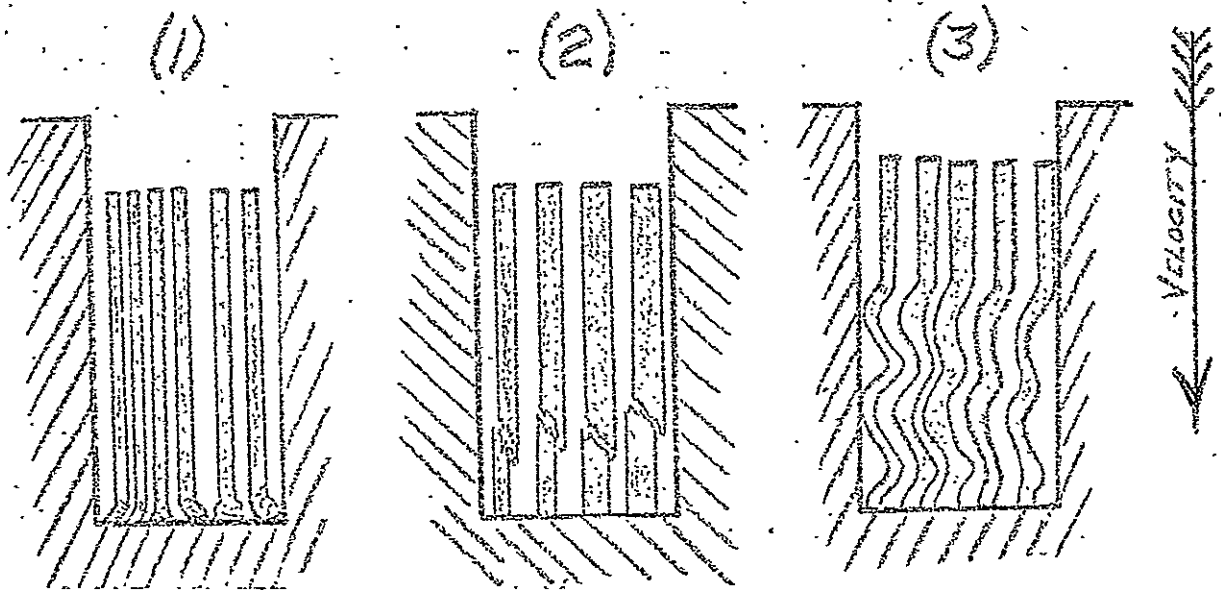
Negative Plate material has been found to have a fracture load of 135 lbs for the 2" width (2400 psi). This is slightly lower than the average value obtained in static testing (recall 168 lbs or 3000 psi). Even recognizing that the static data contained a great deal of scatter, we may assume that some dynamic phenomena are being observed. If our deformation model for compression is valid, it would imply that the Ni metal screen wires are not available at ultimate load to carry significant stress and therefore the strain rate, $\dot{\epsilon}$, effects observed in tensile loading will not appear in compression. In other words elevation of σ_{ult} in tension at elevated $\dot{\epsilon}$ was an observable and reasonable effect but we now have reason to expect very little difference in compressional σ_{ult} as a function of $\dot{\epsilon}$.



4. Mechanical Properties of Electrodes in Bearing Loading

a) Introduction

The determination of the mechanical properties of Ni-Cd electrodes under tensile and compressive loading has been discussed. This section describes the third and final method of loading employed to complete the strength characterization of this material. Bearing tests, although certainly related to the study of compression loading have been distinguished here as a separate method. The main reason is that we did not insist upon any one failure mode. Instead, we attempted to shift from one failure mode to another in a controlled fashion. A second reason for distinguishing this series of tests is that the present manner of loading more closely approximates the type of freedom and failure that might be experienced in an actual battery. It was envisioned that under high gravitational fields our plate material might exhibit any one of three failure modes:



- (1) True Bearing Failure: The bearing failure mode would be exemplified by edge crushing of the plate material at the fixed impact surface. If this failure mode were exhibited, it would occur at a lower value of inertial stress than that maximum required to cause true compressional failure. This failure mode has significant implications to the battery manufacturer since it is most likely to cause separator penetration and shorting of the cell.
- (2) Compression-Shear Failure: The true compression failure mode would be exhibited if ultimate bearing stress exceeded ultimate compression stress. If this is the case, then the value observed would represent the maximum inertial stress that can be supported by this



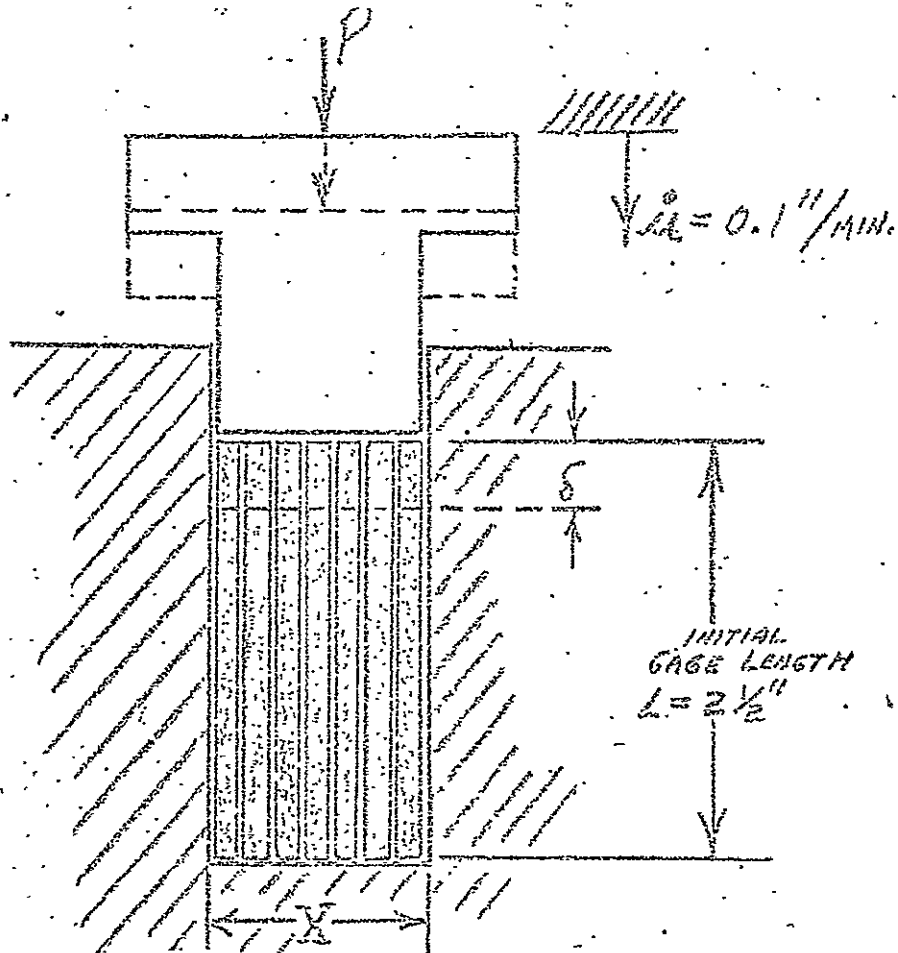
material. It is obvious, therefore, that the battery designer would be extremely interested in this determination.

- (3) Buckling Failure: The buckling mode of failure will be observed at a lower stress level than that associated with either failure mode (1) or (2) and is obviously to be avoided. Hence, the determination of the degree of lateral restraint necessary to prevent this transfer of failure mode is of prime design importance.

With these considerations and goals in mind, the experimental program was outlined to investigate the occurrence of all three failure modes under controlled lateral restraint at both static and dynamic rates of loading.

b) Constant Deformation Static Testing

The static bearing loading was accomplished on a universal testing machine at a constant deformation rate ($\dot{\epsilon} \approx 0.0007 \text{ sec}^{-1}$). The manner of specimen loading is shown on the proceeding page:



The essential characteristic of this loading arrangement was the controlled lateral restraint. The degree of lateral restraint was normalized for all tests to a value of additional space allowed for each electrode.

$$S = \frac{X - [(t_{p1} + t_{p2} + \dots + t_{pn}) + (n + 1)t_s]}{n}$$

S = excess deformation space permitted per plate (in)

X = constant dimension of fixture

t_{p1} = thickness of plate (Typ. 0.026")



t_s = separator thickness

n = number of plates

Thus, stress, σ , could be defined as:

$$\sigma = \frac{P}{w(t_{p1} + t_{p2} + \dots + t_{pn})}$$

where the denominator is simply the area of plates being compressed and P is the total force as recorded from the load cell of the machine on the y-axis. Strain in this study was taken as $\epsilon = \frac{\delta}{L}$ and again $\delta = \mu^0 t$ was directly displayed on the x-axis of our recorder.

The static bearing tests carried out under this type of loading were conducted on positive and negative plates at spacing values ranging from .000"/plate - .009"/plate. The specimens are shown in Figure III-10. The progression from left to right illustrates the complete transfer of failure mode from true compressional-shear to buckling as the spacing is increased. We observe the ultimate loads decrease to that minimum level associated with pure buckling. Important to note is the total absence of edge-crushing or true bearing failure. What is observed then, is an ultimate bearing stress which seems to be greater than the ultimate compression stress for this material



under static conditions.

Ultimate strain levels were fairly uniform regardless of failure mode mainly due to our definition of an overall apparent macro-strain. The stress levels are important however and large differences have been exhibited in direct relation to the spacing allowed and our ability to restrain buckling. The values of buckling stress and ultimate compression stress obtained in this study agree very well with the values determined earlier under strict compression loading as demonstrated in Table III B-7.

TABLE III B-7

<u>Material</u>	<u>Static Buckling Stress</u>		<u>Ultimate Comp. Stress</u>	
	<u>Compression</u> <u>(1/2" G.L.)</u>	<u>Bearing</u> <u>(Max. Spac'g.)</u>	<u>Compression</u> <u>(1/4" G.L.)</u>	<u>Bearing</u> <u>(Min. Spacing)</u>
Pos. Plate	2700	3000	5600	6500
Neg. Plate	2300	2300	3000	4900

c) Dynamic Bearing Loading

Having established a static strength foundation, we were interested in observing possible changes in these values because of increased loading rate. Dynamic bearing tests were performed by use of a loading fixture as shown on the proceeding page:

Figure III - 10

STATIC BEARING

NEGATIVE

MAXIMUM STRESS (S)



.000\"/>

S = 4910 PSI



.001\"/>

S = 4000 PSI



.0025\"/>

S = 3715 PSI



.0045\"/>

S = 2760 PSI



.009\"/>

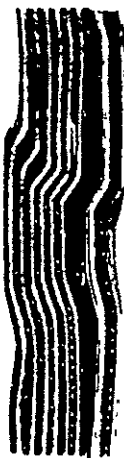
S = 2290 PSI

POSITIVE



.000\"/>

S = 6500 PSI



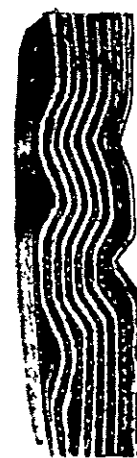
.0025\"/>

S = 4585 PSI



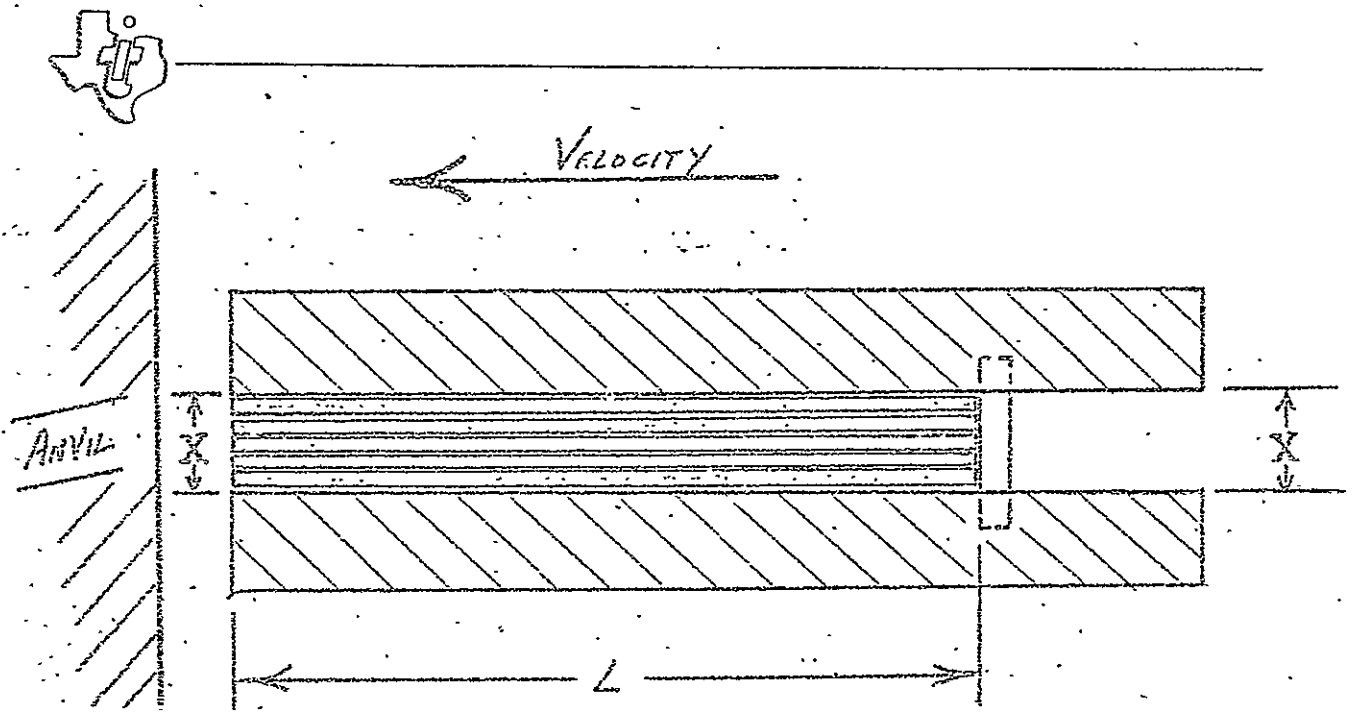
.001\"/>

S = 5480 PSI



.0045\"/>

S = 1385 PSI



In this arrangement the plates were not "fixed" as in the compression fixture but were free to move and to crush at the ends if sufficient force was applied. Controlled spacing/plate, however, through the use of shims, was the essential feature of this fixture in order to duplicate the static conditions of loading. Initial plate length and impact level could be varied and thus maximum stress, σ_{\max} , has been defined here as:

$$\sigma_{\max} = \frac{P}{A} = \frac{m(Ng)}{(t \cdot w)} = \frac{(w \cdot L \cdot \rho)(Ng)}{g(t \cdot w)}$$

$$\sigma_{\max} = \left(\frac{P}{t}\right)(LN)$$

where: ρ = area wt. density (#/in²)

t = plate thickness (in)

L = plate length (in)

N = number of g's accel. (-)

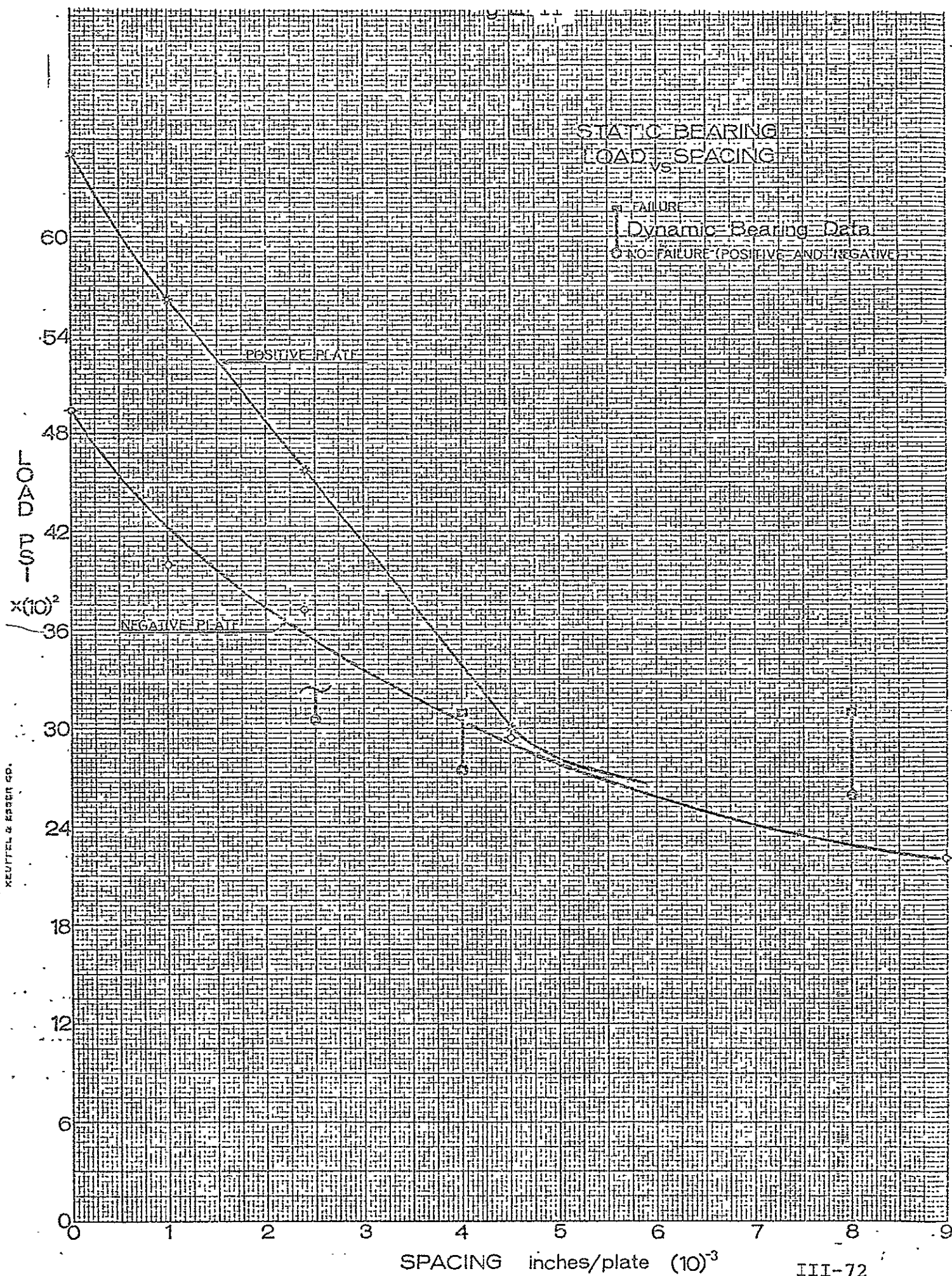


Thus, for a given test at a controlled spacing/plate, the maximum stress would be calculated from the measurement of plate dimensions and applied deceleration. This max. stress would either cause visual failure at the end of the plates or it would not. If this max. stress was sufficient to fail the plates, we would lower the value to find the point where the material would not fail. If the same max. stress was not sufficient to fail the plates, the level would be raised to the point of failure. This partly explains the span lines of experimental determination of failure-non-failure shown in Figure III-11. This procedure was followed for each value of spacing selected. Figure III-11 presents the values of dynamic failure loads superimposed on the static failure loads as a function of plate spacing or degree of lateral restraint. The static curves are well behaved and exhibit the greater strength of the positive electrode and show both electrodes tending toward a nearly common buckling stress asymptote. This is again in agreement with the earlier static compression study. These curves display the need to maintain a high degree of lateral restraint if excessive loading must be endured.



The dynamic load levels that are indicated as span lines on Figure III-11 are not solely the limits of two experimental levels. We have included in these lines the results of both positive and negative plates. Several of each type were placed in each fixture and the difference in stress was very small. This was essentially as expected since it agreed with the static results observed -- that at larger spacing, buckling is dominant and there is thus little distinction between the two electrodes. This is shown by the joining of the two static curves and their tending toward a common asymptote.

As lateral restraint was increased, however, we reached a point where we were unable to produce failure in either electrode (single point on Figure III-11 @ .0025" spacing). In other words, taking the longest plate (6-5/8") that could be accommodated in our fixture, and employing the highest impact level (meas. mean 4000g) for 1.1 msec from 110 ft/sec). Available at that time, the maximum stress developed (3100 psi for NEG.; 2900 psi for POS.) was not sufficient to produce failure. Thus, only a lower bound could be established at this degree of lateral restraint (see Table III B-8).





It appears, however, that the dynamic curve, if able to be completed, would have a shape or character similar to the static curve. The dynamic values of ultimate stress are nearly equal to those obtained statically and reinforces our position that σ effects may not be as important in compression. This is valuable information in that it makes the transfer from static to dynamic results less complicated.

The failure modes exhibited in the dynamic study included all three mechanisms to some degree. In the case of minimum lateral restraint, the onset of buckling at the impact end was dominant as expected. As we increased the degree of lateral restraint, both compression-shear (very close to the impact edge) and bearing failure (edge crushing of the impact surface) were displayed in nearly equal amounts. It appears, therefore, that in high rate loading the ultimate bearing stress nearly coincides with the ultimate compression stress.

It is valuable as a final summary to tabulate the dynamic results obtained in this study and compare them with previous determinations of dynamic ultimate strength.



TABLE III B-8

	Dynamic σ_{ult} Tension	Dynamic σ_{ult} Compression	Dynamic σ_{ult} @.008"	Bearing σ_{ult} @.004"	@.0025"
Pos. Plate	2000	>2600	2800	2800	>2900
Neg. Plate	2000	2400	3000	2800	>3100

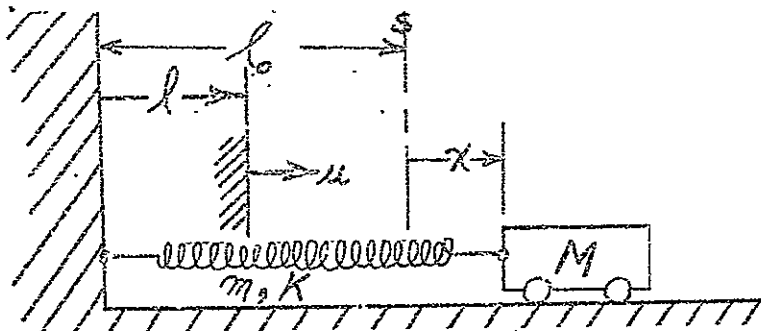
C. DYNAMIC RESPONSE OF ELEMENTS TO IMPACT LOADING

1. Theoretical Predictions of Dynamic Response

a) One Degree of Freedom Analysis of Natural Frequency:

The natural frequencies, f_{N_n} , of a vibrating system or element are a most important parameter in the study of the transient as well as the steady state response of that system. Therefore, our preliminary analysis effort was directed toward a modified single degree of freedom approach to approximate the natural frequencies of our system. With this approximation, we could then examine the consequences of the value of f_{N_1} so obtained with respect to the shock spectra of the specified deceleration pulse.

The single degree of freedom system employed is shown below:



l_0 = unstressed length of spring
 S = static equilibrium position
 x = dynamic displacement of body
 M = mass of body

l = length along spring
 u = dynamic displacement of dl
 m = mass of spring
 K = spring constant

This system permits the mass of the spring to be considered and $\omega_{N_1} \approx \sqrt{\frac{K}{M}}$ as in the simple case. Thus we can use this system to reduce our distributed-mass (infinite degree-of-freedom) element (battery electrode) to a lumped-parameter system utilizing the concept of vibrational transfer of potential and kinetic energy. This method has been accurately described in the literature and for this system it has been shown that the natural frequency (undamped) is:

$$\omega_{N_1} = \frac{\pi}{2} \sqrt{\frac{K}{2M+m}} \quad \text{for } M \ll m$$

(Rad./Sec.)



Thus if $M \rightarrow 0$, we have the case of a spring alone without concentrated end mass and:

$$\omega_{N_1} \approx \frac{\pi}{2} \sqrt{\frac{K}{m}} \quad \text{for } M = 0$$

or for our elastic spring element of length, l , apparent cross-sectional area, A , and apparent modulus of elasticity, E , we have:

$$f_{N_1} = \frac{\omega_{N_1}}{2\pi} \approx \frac{1}{4} \sqrt{\frac{AE}{lm}} \quad (\text{cy/sec})$$

Now for battery electrodes we may write for convenience:

A = width x thickness = wt

$$m = \frac{\text{weight}}{g} = \frac{\text{length} \times \text{width} \times \text{area density}}{g} = \frac{l \cdot w \cdot \rho}{g}$$

where: ρ = area weight density

$$f_{N_1} = C \sqrt{\frac{E}{l}} \quad \text{Where: } C = \frac{1}{4} \sqrt{\frac{t \cdot g}{\rho}}$$

Specifically for our Ni-Cd electrodes:

t = thickness (.025"-.028") = 0.0265 in.

g = acceleration of gravity = 386 in/sec²

ρ = area weight density = 3.2×10^{-3} #/in²

(2.8-3.2 $\times 10^{-3}$ #/in² positive)

(3.3-3.5 $\times 10^{-3}$ #/in² negative)

Variables:

E = modulus of elasticity

0.1×10^6 psi negative plate in compression

1.0×10^6 psi positive plate in tension



l = length of electrode

normally 3" - 8"

Thus: $C = 14$

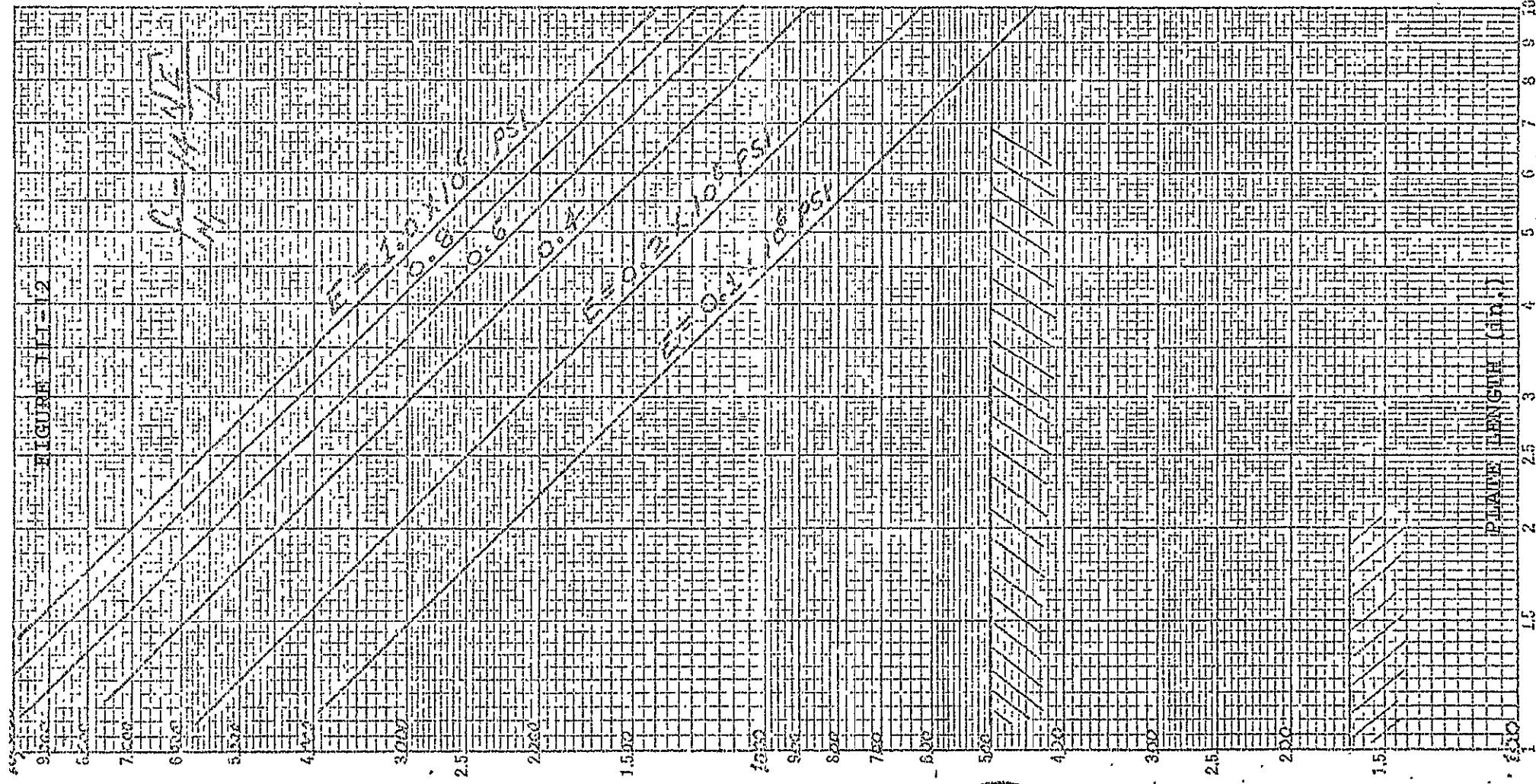
and the fundamental frequency of longitudinal vibration for sintered Ni-Cd electrodes is given by:

$$f_{N_1} = (14) \sqrt{\frac{E}{l}}$$

This fundamental natural frequency has been calculated for various plate lengths and values of E and is plotted in Figure III-12.

b) Shock Spectra Implications:

A shock spectrum is a plot showing the peak response of a variable-frequency, single degree of freedom oscillator to a specific shock pulse as a function of the natural frequency of the oscillator. (It is important to understand from the start, that the shock spectrum tells us what a given shock pulse will do; not what it is.) This is much different from a Fourier spectrum or plot of the Fourier analysis performed to describe a given shock pulse. It has been found that when a single degree of freedom, linear, undamped system with a distinct natural frequency is subjected to a mechanical shock, its response motion will be determined by the magnitudes, shape, and time duration of the shock pulse. In fact, the amplification spectrum as it is sometimes called, is



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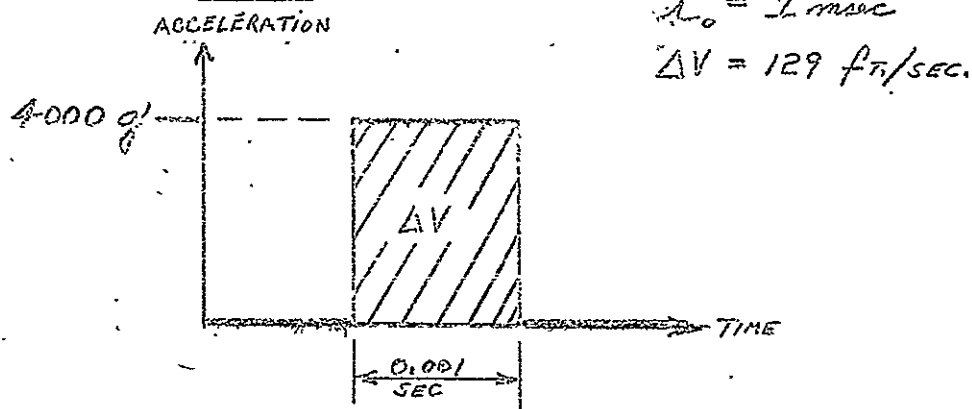
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rather insensitive to the exact time-history of the pulse. Rather, the rise-time of a given pulse is the most important factor which distinguishes the maximum amplification factors for various pulse shapes. The total impulse and rise time are the two main damage criteria. As will be seen, the rapid rise time of our square pulse implies the highest maximum value of any pulse shape (theoretically = $2X$). In general, the shock spectrum will not have frequency alone as the abscissa because of the above considerations, rather a ratio of the shock pulse length (t_0) to the natural period of the oscillator (T) is more useful $\frac{(t_0)}{T}$. Depending on the value of this ratio, however, we find that the system's response during application of a given shock pulse may differ considerably from the motion after the shock pulse has ended ($t = t_0$). It is therefore common to plot a "primary" spectrum (response during shock input $0 < t < t_0$) and a "residual" spectrum (response after shock input $t_0 < t < \infty$) to fully generalize the behavior of various elements to various shock durations of a certain pulse shape. However, it is more useful to provide the designer of shock-resistant elements with a single "maximax" spectrum plot arrived at by taking the greater value from the "primary" or the "residual" plots throughout the frequency ratio range of interest.



It is not appropriate to reiterate the derivations and analyses of various spectra characteristics that can be found in the literature. The value of using this concept is well recognized and it is only important that we understand the fundamentals in order to apply the information that can be obtained. Thus, the character of the maximax shock spectra can be obtained directly from the literature. In our case, we wish to consider a single shock input:



Thus, we can establish the abscissa directly as the natural frequency of our Ni-Cd electrodes rather than as a time ratio.

The ordinate is simply the relative response and can be scaled directly in numerical dynamic amplification factors. It must be remembered that the response can be thought of as displacement, acceleration, etc. (If this same quantity describes the input pulse). More useful to the designer is the concept of force or stress (indirectly strain) amplification over that of a



static loading equal to the amount of input acceleration.

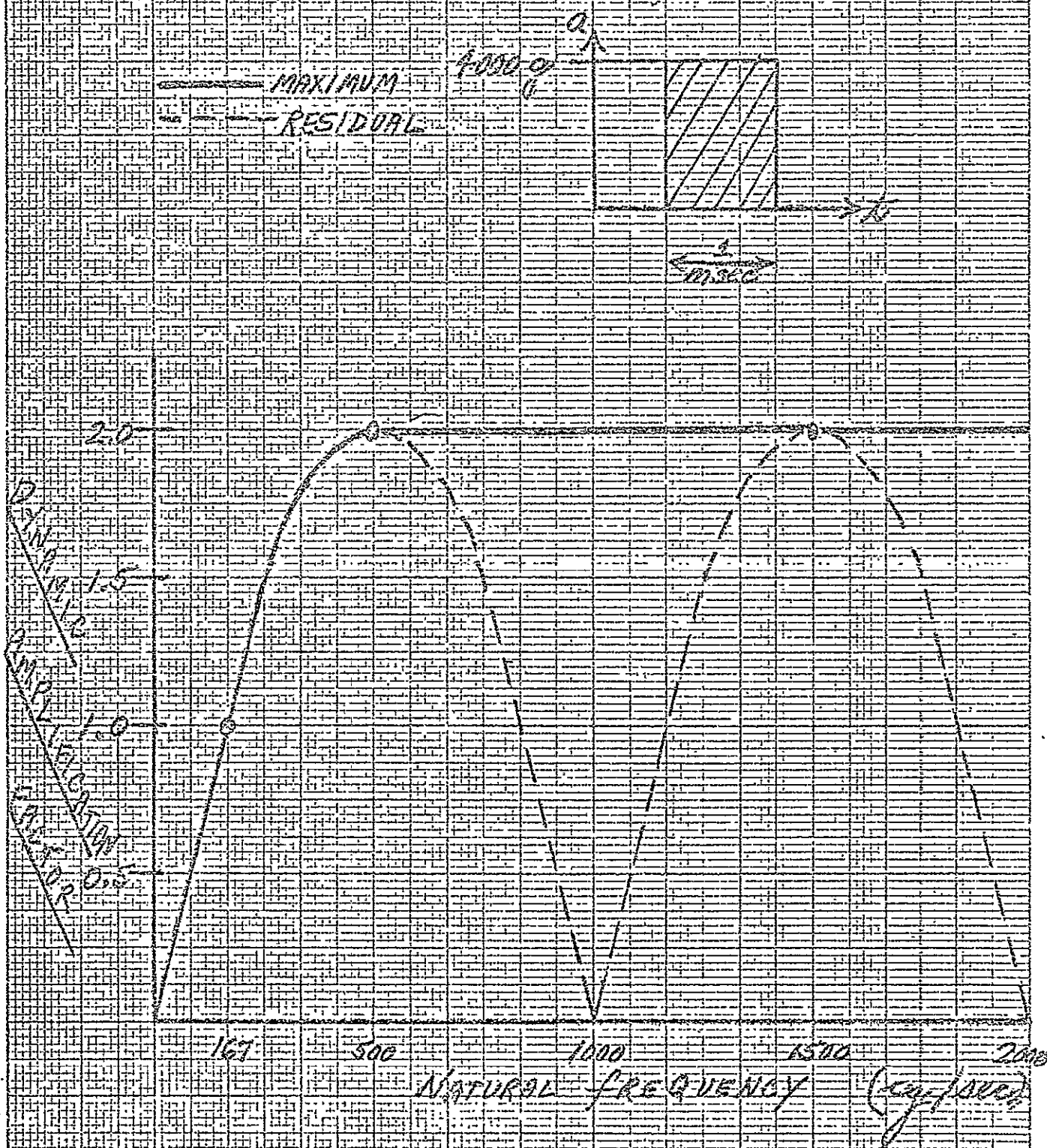
The shock spectrum for the pulse illustrated above is given in Figure III-13 with the ordinate and abscissa scaled as discussed for optimum use in our problem. The basic implications are: (1) Any Ni-Cd electrode with a natural frequency greater than 167 cy/sec will experience an amplification of the nominal input forces under dynamic loading. (2) Any Ni-Cd electrode with a natural frequency greater than 500 cy/sec will experience twice the nominal input forces under this dynamic loading situation.

From a design point of view, the calculation of natural frequencies in the previous section now has more value. It may be instructive to outline the design considerations that have been formulated:

- (1) The manner of loading must first be identified and this will dictate the modulus of elasticity and strain-rate effects from previous determinations of the mechanical properties of our electrode.
- (2) Space considerations may indicate a trial size of our electrode.
- (3) The design pulse magnitude will be used to calculate the static stress applied to this size and weight electrode.

FIGURE III-13

SHOCK SPECTRA FOR PULSE





- (4) The natural frequency of this specific electrode is determined.
- (5) The shock spectrum (for the design pulse) is used to determine the dynamic amplification factor which applies.
- (6) The actual stress which this electrode will experience under this design pulse loading can now be calculated.
- (7) The effect of this stress on the electrode is now determined from previous determinations of this electrode's mechanical properties under the same manner of loading.

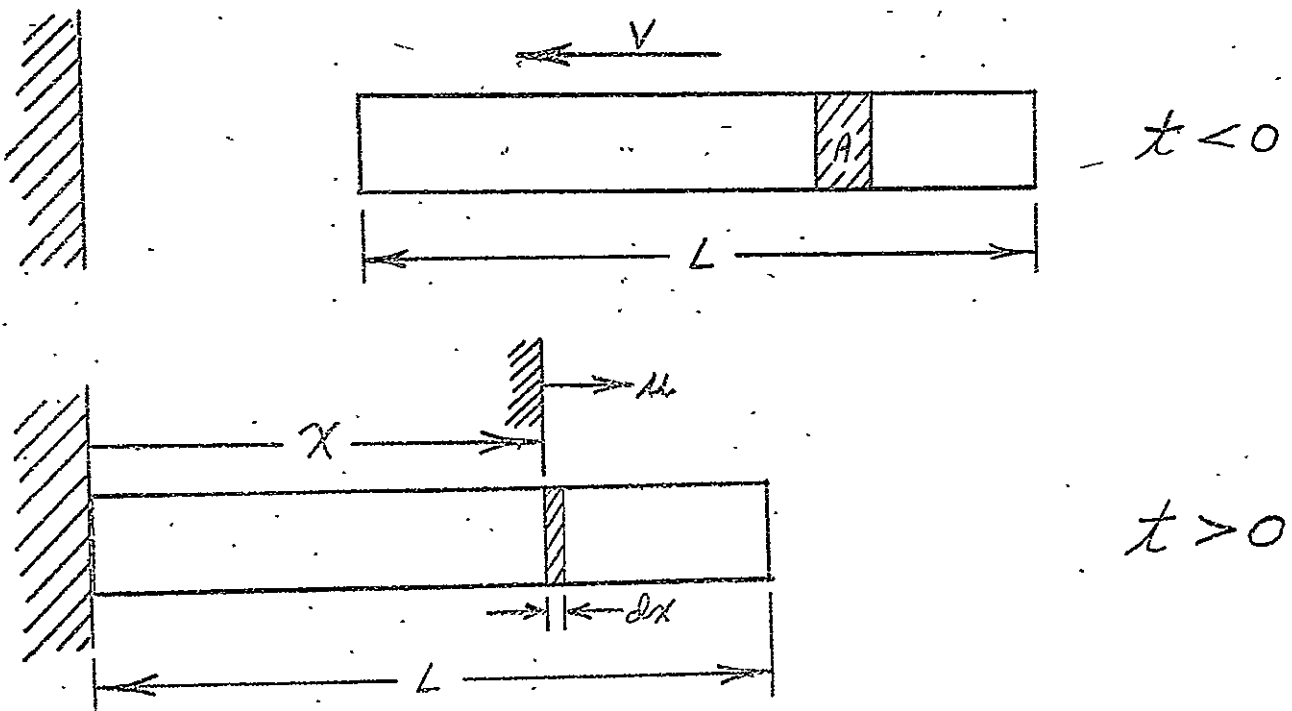
The importance of determining complete experimental information concerning the mechanical properties of Ni-Cd electrodes is now obvious. Without such information, design can neither begin (Step 2) nor can a valid judgment be made (Step 7) concerning the actual damage that may or may not occur to an electrode in a specific loading situation.

c) Analysis of Displacement and Stress for Distributed-Mass Element:

The frequency-shock spectra analysis is quite straightforward and can provide valuable information to the designer. It does not permit a direct calculation of the stress within the element, however, and it does not



give us a description of the deformation process. What we would like is an insight into the actual displacements within an electrode and the stresses that are applied during impact. We seek an accurate description of stress and strain as a function of location and time. This type of information can only be obtained by a more rigorous analysis of our distributed-mass element. To explore the possibility of solution, we have formulated the problem as a long, slender element of uniform cross-sectional area, A , and constant initial velocity (rigid body motion), $-V$, which is subjected to a sudden longitudinal deceleration at time $t = 0$. The problem is shown below:





We find that the governing differential equation is:

$$\frac{\partial^2 u}{\partial t^2} = a^2 \frac{\partial^2 u}{\partial x^2} \quad (2)$$

which is the general "wave equation" in which:

$$a = \sqrt{\frac{E}{\rho}} \quad \text{where: } \rho = \text{mass density}$$

We have for our case:

Initial Conditions	Boundary Conditions
$u(x, 0) = 0$	$u(0, t) = 0$
$\frac{\partial u}{\partial t}(x, 0) = -v$	$\frac{\partial u}{\partial x}(L, t) = 0$

imposed by our considerations of constant initial velocity, rigid fixed end and free unstressed end.

Following the method of separation of variables we assume the general solution of EQ. (2) may have the form:

$$u(x, t) = X(x) T(t) \quad (3)$$

Now we may write EQ. (2) as:

$$X T'' = a^2 T X''$$
$$\frac{T''}{T} = a^2 \frac{X''}{X} = -\lambda^2$$

the component ordinary differential equations will be:

$$T'' = -\lambda^2 T \quad \& \quad X'' = -\frac{\lambda^2}{a^2} X$$

and whose solutions are known:

$$T = A \cos \lambda t + B \sin \lambda t$$

&

$$X = C \cos \frac{\lambda}{a} x + D \sin \frac{\lambda}{a} x$$



Therefore, the general product solution EQ.3 can be written:

$$u(x,t) = (C \cos \frac{\lambda}{a} x + D \sin \frac{\lambda}{a} x) (A \cos \lambda t + B \sin \lambda t) \quad (4)$$

Examining the first boundary condition in EQ.(4):

$$\Rightarrow C = 0 \text{ in general for all } t$$

Examining the second boundary condition in EQ.(4):

$$\Rightarrow (D \frac{\lambda}{a}) \cos \frac{\lambda}{a} L = 0 \text{ in general for all } t$$

$$\text{Hence: } \lambda_n = \frac{(2n-1) a \pi}{2L} \quad n = 1, 2, 3, \dots$$

Now the general solution EQ.(4) has been reduced to a series of product solutions:

$$u(x,t) = \sum_{n=1}^{\infty} \sin \frac{(2n-1)\pi}{2L} x \left[A_n \cos \frac{(2n-1)\pi a}{2L} t + B_n \sin \frac{(2n-1)\pi a}{2L} t \right] \quad (5)$$

Examining the first initial condition in EQ.(5):

$$\Rightarrow A_n = 0 \text{ in general for all } x$$

Examining the second initial condition in EQ.(5):

$$\left. \frac{\partial u}{\partial t} \right|_{t=0} = \sum_{n=1}^{\infty} B_n \left(\frac{(2n-1)\pi a}{2L} \right) \sin \frac{(2n-1)\pi}{2L} x = -V \quad (6)$$

We recognize a similarity between EQ.(6) and a half-range Fourier Series expansion. Hence, the problem of determining B_n in EQ.(6) is really the problem of expanding the given function $f(x)$ in a half-range sine series expansion over the interval $0 < x < L$.

Performing the required expansion and integration to determine the coefficients we find:



$$B_n = \left[\frac{-8 VL}{(2n-1)^2 \pi^2 a} \right] \quad (8)$$

Now we may write the final general equation for displacement from EQ. (5):

$$u(x,t) = \frac{-8VL}{a \pi^2} \sum_{n=1}^{\infty} \frac{1}{(2n-1)^2} \sin \frac{(2n-1)}{2} \pi \frac{x}{L} \sin \frac{(2n-1)}{2} \pi \frac{at}{L} \quad (9)$$

which fully describes the elastic displacement at any time and any longitudinal location along the electrode. The natural frequencies of this sinusoidal displacement $u(t)$ or vibration are:

$$f_{N_n} = \frac{\lambda n}{2\pi} \quad (\text{cycles/sec}) \quad n = 1, 2, 3, \dots$$

therefore:

$$f_N = 1 \left(\frac{a}{4L} \right), 3 \left(\frac{a}{4L} \right), 5 \left(\frac{a}{4L} \right), \dots \quad (10)$$

Examining the fundamental natural frequency:

$$f_{N_1} = \frac{a}{4L} = \frac{\sqrt{E/\rho}}{4L} = \frac{\sqrt{Eg/\gamma}}{4L} = \frac{1}{4} \sqrt{\frac{tg}{\rho}} \frac{\sqrt{E}}{L}$$

We find that it is equivalent to the "approximation" we had obtained earlier in Section III C.1.a. in our consideration of the single degree of freedom system. This is not actually surprising since the earlier approximation employed sine functions as "guesses" for both the normal function and the time function. This means now that Figure III-12 gives us the true natural frequency, f_{N_1} , as confirmed by a rigorous analysis.



Having obtained a complete solution (EQ.(9)) for elastic displacement, we can now investigate the general stress solution:

$$\sigma = E \epsilon = E \frac{\partial u(x,t)}{\partial x}$$

If this initial assumption of elastic material behavior is retained, the general stress solution is:

$$\sigma(x,t) = \frac{-4EV}{a\pi} \sum_{n=1}^{\infty} \frac{1}{(2n-1)} \cos \frac{(2n-1)}{2} \pi \frac{x}{L} \sin \frac{(2n-1)}{2} \pi \frac{at}{L} \quad (11)$$

which fully describes the elastic stress within the element at any time and any longitudinal location along the electrode.

We are interested in the expression for maximum stress which occurs at the fixed end ($x = 0$):

$$\sigma_{\max.}(0,t) = \frac{-4EV}{a\pi} \sum_{n=1}^{\infty} \frac{1}{(2n-1)} \sin \frac{(2n-1)}{2} \pi \frac{at}{L} \quad (12)$$

Numerical Analysis:

We have attempted to numerically evaluate the maximum stress (EQ.(12)) for a few values of time, t . It is apparent that we have simply the summation of the fundamental and all the odd sinusoidal harmonics which is of course a square wave. The coefficients of the odd harmonics appear to imply a $\frac{\pi}{4}$ multiplier for the fundamental. At the fixed end where stress is a maximum,



we have a state of compression for the time interval, $0 < t < \frac{2L}{a}$, followed by a state of tension for the time interval $\frac{2L}{a} < t < \frac{4L}{a}$. This process is then cyclic at a frequency of $(\frac{a}{4L})$ cycles/sec.

Conclusion:

The sole limitation of the general solution obtained in this section is the one assumption that made the solution possible -- elastic material behavior. If we do not impose this material limitation and allow stress to be some general function of strain, we do not obtain EQ.(2) as the governing differential equation.

Rather:

$$\frac{\partial \sigma}{\partial \epsilon} \left(\frac{\partial^2 u}{\partial x^2} \right) = \rho \frac{\partial^2 u}{\partial t^2} \quad (13)$$

whose general solution is unknown. EQ.(13) therefore introduces the most significant portion of non-linearity into our system. We understand that JPL has devoted some effort to the treatment of this and large displacement non-linearities. Recent literature has discussed ways of introducing empirically the non-linearities implied by plastic and strain hardening behavior. This area is in need of further analysis. The research indicated for a more complete understanding of dynamic response is in the introduction of these material



behavior non-linearities into the general governing differential equation (13) and the physical interpretation of its solution with the aid of computer analysis.

One further point does not concern the validity of the solution given, but concerns the exactness of its applicability to our experimental loading situation. In question is the first boundary condition which was used in our solution to describe the rigid fixed end.

We should go back and examine the effect on our solution of imposing a condition of uniform deceleration during the time interval of impact:

$$u(0,t) = -vt - \frac{a}{2} t^2 \quad 0 < t < .0001/\text{sec.}$$

This more accurately describes the displacement of this end of the bar during the time of loading and could thus make the existing solution conservative in its prediction of stresses which may be higher than those actually experienced in our testing.

2. Experimental Determinations of Dynamic Reponse

a) High Speed Photographic Analysis

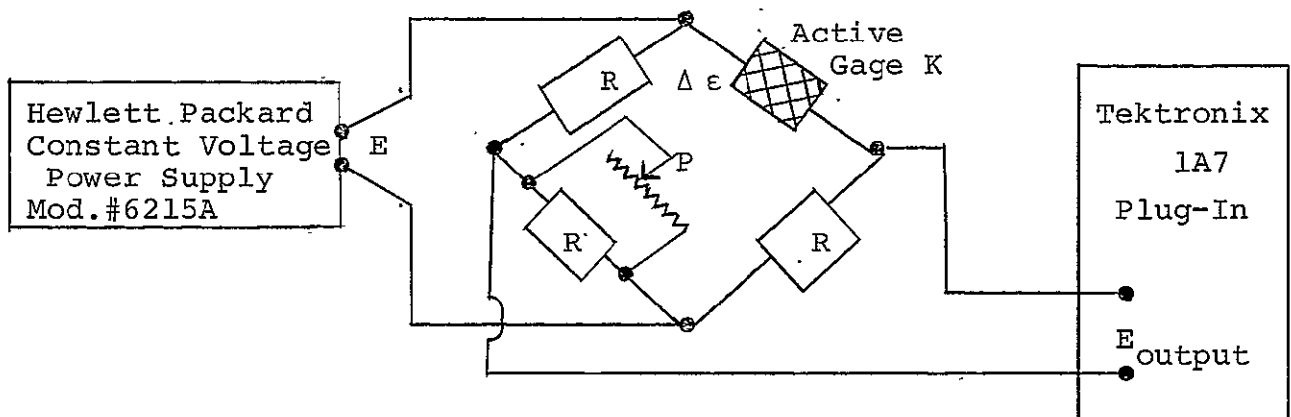
Following an analysis or theoretical prediction of mechanical behavior, we most desired an experimental verification of that prediction. One method to experimentally obtain the actual displacements of an element



during a loading situation is to visually observe and measure them. We have, therefore, used high speed photography to examine the motion of our electrodes during impact. It appears that a great deal of information concerning the description of dynamic displacements can be gained by this method.

b) Dynamic Strain Measurements

Some work has been done in the measurement of dynamic strain under impact conditions using the strain-gage techniques we have developed. The basic circuitry and instrumentation system was simply a wheatstone bridge employing a single active strain gage with zero balance:



K = Active Gage Factor (2.0)
 R = Fixed Resistor (120 Ω)
 P = Zero Balance Potentiometer
 E = 12 Volt Supply

$$E_{\text{output}} = \frac{R^2 K E}{(R+R)^2} (\Delta \epsilon) = 6.0 (\Delta \epsilon)$$

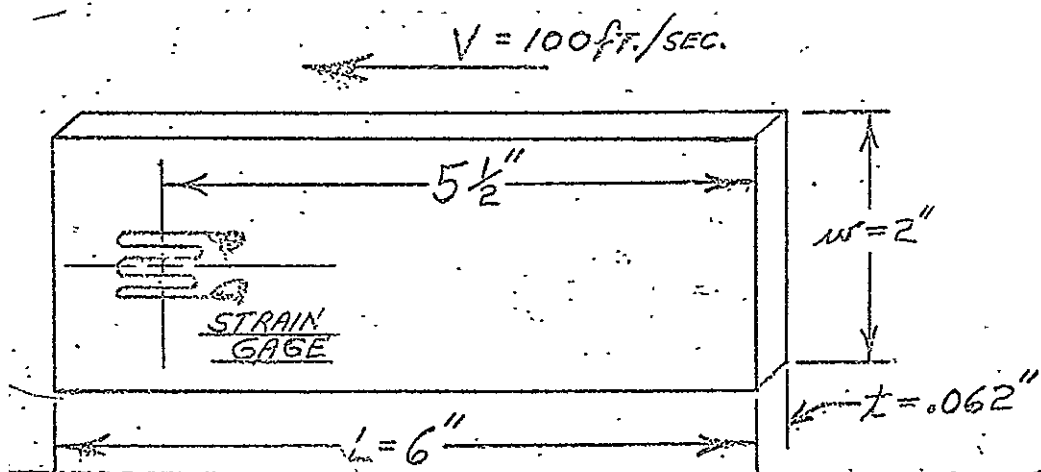
This circuit was used for all the experimental data contained in this report. Some work was done near the end of the experimental program involving two active strain gages to explore the dynamic strain distribution along the plate length, $u = u(X, t)$. This data was very limited, however, and rather inconclusive. Some correlation with our theoretical predictions of strain distri-



bution with time, $u = u(x, T)$, was indicated before the experimental work ended. This portion of the report indicates the relation of these preliminary findings to the theoretical predictions that have been made.

(1) Aluminum Plates:

The mechanical properties of the aluminum alloys are well established and readily available. An aluminum plate loaded within its elastic region is obviously a more well-behaved system than our porous plate material. Thus, our initial experimental work was aimed at confirming our theoretical analysis by the use of an aluminum plate:



6061 - T6 Alloy:

$E = 10.0 \times 10^6$ psi
 $\sigma_y = 40,000$ psi
 $\sigma_{ult} = 45,000$ psi
 $\gamma = 0.0976$ #/in³
 $\rho = 0.00605$ #/in²
 $a = 16,600$ ft/sec

Single Strain Gage:

$R = 120 \Omega$
 $K = 2.0$
 $E = 12.0$ Volts
 $\Delta E = \frac{R^2 K E (\Delta \epsilon)}{(R \& R)^2} = (6.0) \Delta \epsilon$

We could first calculate the natural frequency, f_{n_1} , of this element by either of the two equivalent equations we have given earlier:

$$f_{n_1} = \frac{1}{4} \sqrt{\frac{tg}{l}} \sqrt{\frac{E}{L}}$$

Figure III - 14

UPPER TRACE:

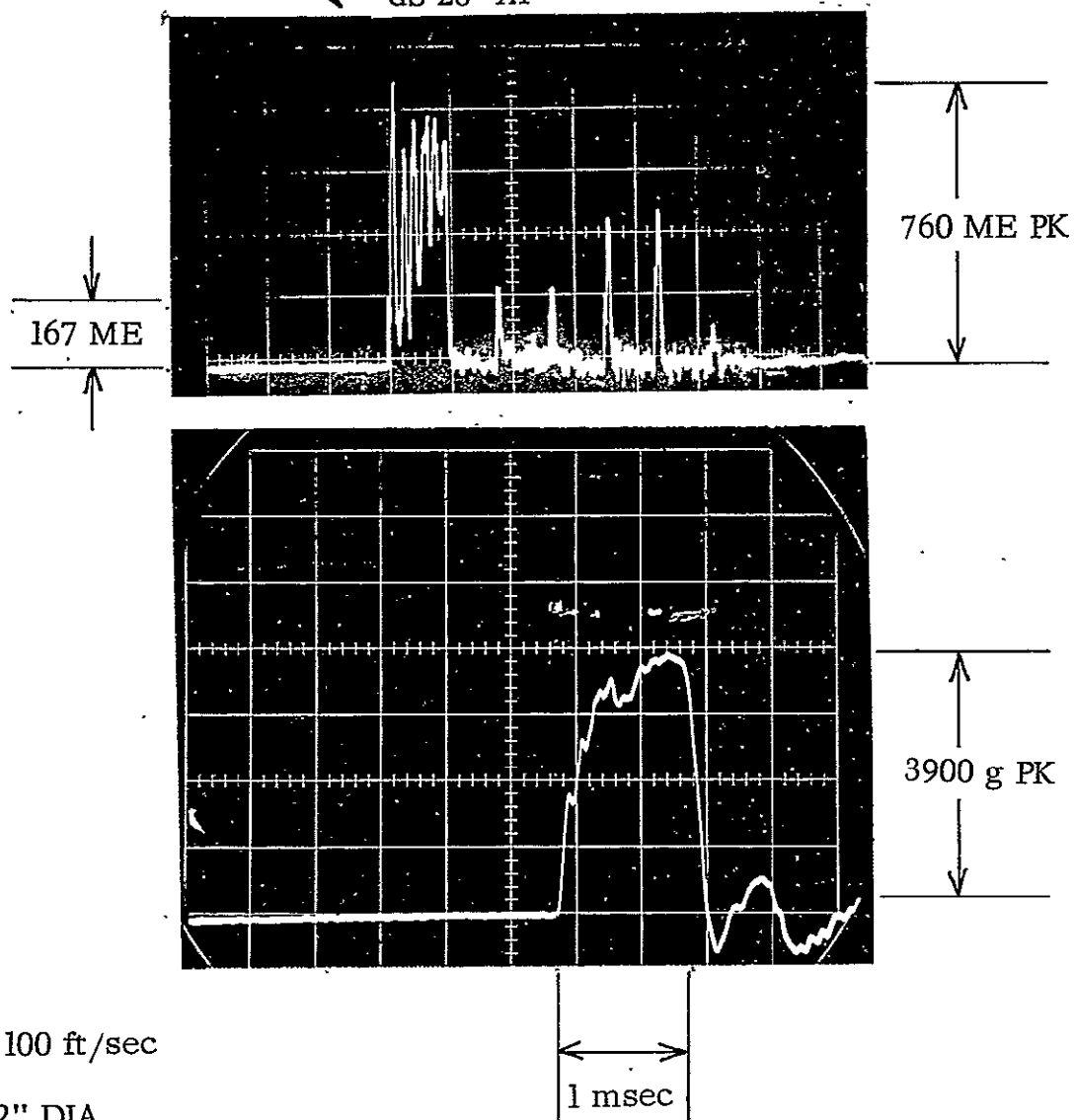
DYNAMIC STRAIN - 1 msec/cm; 1 mo/cm

LOWER TRACE:

ACCELERATION PULSE - 0.5 msec/cm; 1000 g/cm

6,500 cy/sec

dS 28 Al



V = 100 ft/sec

1 1/2" DIA.

DYNAMIC STRAIN OSCILLOGRAPH OF AL PLATE



$$\text{or } f_{n_1} = 1 \left(\frac{a}{4L} \right)$$

and we find $f_{n_1} = 8,300$ cy/sec.

The lower trace of Figure III-14 shows the acceleration time history of loading imposed on this plate during impact. We note that the pulse length (t_0) is 0.001 sec. The shock spectrum given in Figure III-13, therefore, applies directly. Since the calculated $f_{n_1} = 8,300$ cy/sec $\gg 500$ cy/sec, we can only conclude from this spectrum that a dynamic amplification factor equal to 2.0 must be used.

Due to simple inertia loading, the static stress level ($L=6"$) may be calculated:

$$\sigma = \frac{ma}{A} = \frac{(L' \cdot w \cdot t \cdot \gamma)}{g (w \cdot t)} (Ng) = L' \gamma N \Rightarrow \sigma_{\text{static}} = 2300 \text{ psi.}$$

Employing the dynamic amplification:

$$\sigma_{\text{dyn.}} = 4600 \text{ psi}$$

$$\text{And: } \epsilon_{\text{dyn.}} = \frac{\sigma_{\text{dyn.}}}{E} = 460 \mu \epsilon$$

is the expected gage output under this loading condition. We now note that the upper trace of Figure III-14 has recorded an experimental peak strain of $\epsilon = 760 \mu \text{ in/in.}$ In the experimental record, it is possible that the subsequent peaks which indicate a more uniform $668 \mu \text{ in/in}$ peak strain level are truly more significant. The difference between measured and calculated strain is then $210 \mu \text{ in/in.}$ This is quite encouraging as a very preliminary experimental result. The predicted natural frequency (f_n) of the cyclic displacement within our element was 8,300 cy/sec and should be recorded by the strain gage. The upper trace of Figure III-14 shows this cyclic strain behavior within the plate at a frequency of approximately 6,500 cy/sec. Again, it is encouraging to see this preliminary correlation but we cannot account for the discrepancy



(1800) cy/sec) at this time. We may now look at the experimental results and the predictions from our detailed analysis. It appears that our series in the max. stress equation (12) converges to $\frac{\pi}{4}$. We may, therefore, calculate the max. elastic stress predicted:

$$\sigma_{\text{max.}}(0,t) = \frac{-4EV}{a\pi} \left(\frac{\pi}{4}\right) = \frac{-EV}{a} = -60,000 \text{ psi}$$

Now:

$$60,000 \text{ psi} > 40,000 \text{ psi} = \sigma_{\text{yield}}$$

In fact not only plastic deformation is implied but $60,000 \text{ psi} > \sigma_{\text{ult}}$!

Even if we assume that a high strain-rate sensitivity elevates the dynamic yield stress and the elastic modulus still applies we have:

$$\epsilon_{\text{predicted}} = \frac{\sigma}{E} = 6,000 \mu \epsilon$$

This value is $>0.2\%$ ϵ normally taken as the yield point; yet there was no observation of plastic deformation after repeated loading at this impact level. Furthermore, the experimental strain trace showed no residual strain (plastic or lack of zero return); yet the fact remains:

$$\begin{array}{lcl} 760 \mu \epsilon \text{ PK} & & 6,000 \mu \epsilon \text{ PK} \\ \text{Experimental} & << & \text{Predicted} \end{array}$$



Our only conclusion must be that our analytical prediction is nearly an order of magnitude conservative! The fault must lie in the first boundary condition we have assumed:

$$u(0,t) = 0 \text{ implies no carriage displacement during the loading period}$$

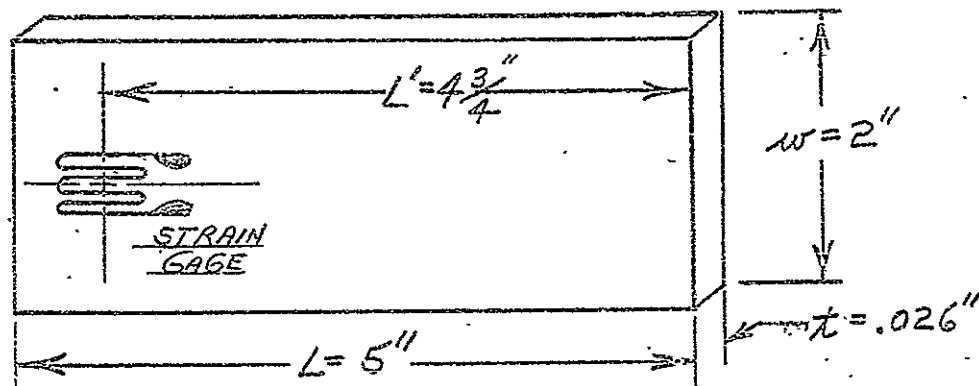
What we really have is the situation that the carriage and the end of the element have an absolute displacement during the loading period. Thus, we must modify our analytical prediction to account for this boundary condition. Consideration of this, of course, will reduce the predicted magnitude and have a strong effect on the discrepancy that has been brought out by our preliminary experimental results. Time did not permit the introduction of this aspect during the current investigation.

(2) Cadmium Electrodes:

Following our study of the aluminum plate, we have obtained some preliminary results on the negative electrode.



$$\leftarrow V = 64 \text{ ft./sec.}$$



Negative Cd Plate

$E_{\text{comp.}} \approx 0.5 \times 10^6 \text{ psi}$
 $\sigma_{\text{ult.}} \approx 3000 - 5000 \text{ psi}$
 $= 0.127 \text{ \#/in}^3$
 $= 0.0033 \text{ \#/in}^2$
 $a \approx .3250 \text{ ft/sec}$

Single Strain Gage

AF-7-1-56 (E-910)
 $\Delta E = (6.0) \Delta E$

Again, it is first useful to determine the natural frequency of this element either by calculation as before or directly from Figure III-12.

$$f_{n1} \approx 2000 \text{ cy/sec}$$

The upper trace of Figure III-15 shows a cyclic strain behavior within the electrode at a frequency of approximately 2500 cy/sec. This is in good agreement with our predicted 2000 cy/sec. Furthermore, it implies that the governing elastic modulus may be higher:

$$E = \left(\frac{f_n L}{14} \right)^2 = 0.8 \times 10^6 \text{ psi}$$

The lower trace of Figure III-15 shows the acceleration-time history of the loading and the pulse length

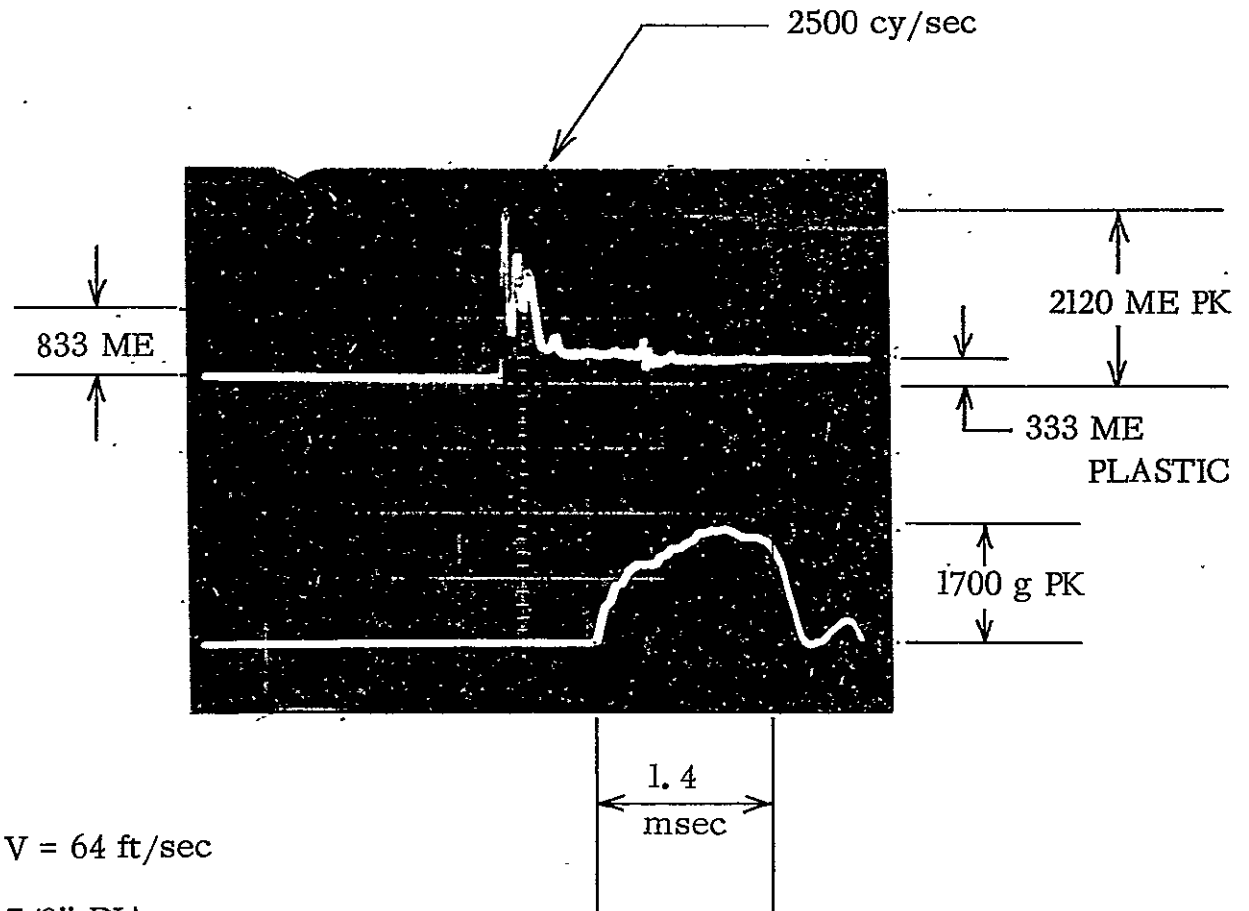
Figure III - 15

UPPER TRACE:

DYNAMIC STRAIN - 2 msec/cm; 5 mo/cm

LOWER TRACE:

ACCELERATION PULSE - 0.5 msec/cm; 1000 g/cm



$V = 64 \text{ ft/sec}$

$7/8'' \text{ DIA.}$

$d = 0.367''$

DYNAMIC STRAIN OSCILLOGRAPH OF Cd ELECTRODE



t_0 is approximately 1.4 msec. Thus, we cannot use the shock spectrum of Figure III-13 directly ($t_0 \approx .001$ sec). However, the amplification factor of 2.0 will apply to any element with a $f_{n1} > 356$ cy/sec for this loading pulse if we assume it is square. It is obvious that $f_{n1} = 2000$ cy/sec $\gg 356$ cy/sec and the maximum amplification factor will apply. The maximum will apply for any shape pulse of this duration. However, we could question the use of the square wave for this low velocity impact level and assume the experimental record more closely approximates a half-sine pulse. If this is true, a maximum amplification factor of 1.8 will apply.

We may now calculate the maximum stress due to simple inertia loading:

$$\sigma = \frac{ma}{A} = L\gamma N = 1080 \text{ psi}$$

Employing the 1.8 x amplification factor:

$$\sigma_{\text{dyn.}} = 1940 \text{ psi}$$

$$\text{which implies: } \epsilon = \frac{\sigma_{\text{dyn.}}}{E=0.8 \times 10^6} = 2430\mu \text{ in/in PK.}$$

The upper trace of Figure III-15 has recorded an experimental peak strain level of only 2120 μ in/in. There is one consideration that might help explain this discrepancy although we have no experimental confirmation at this time. We know that if a material is one-half critically damped ($c'/c = 0.5$), its maximum amplification



factor for a half-sine pulse is only 1.1. If this factor is taken into account we would predict a peak strain of only 1470 μ in/in. This is lower than the recorded peak yet equal to the approximately 1450 μ in/in strain level recorded by subsequent peaks.

This is important enough that it should be investigated if work were continued since it is a very beneficial material property of this porous electrode.

It is important to note the lack of zero-return in the upper trace of Figure III-15. This indicates a residual plastic strain of 333 μ in/in.

Considering the complete analysis, we could evaluate the stress by equation (12) as we did for aluminum:

$$\sigma_{\max.}(o,t) = \frac{-4EV}{a\pi} \left(\frac{\pi}{4}\right) = 9850 \text{ psi}$$

If our original values $E = 0.5 \times 10^6$ psi and $a = .3250$ ft/sec are used for our 64 ft/sec impact. This value of $\sigma_{\max.} \gg \sigma_{\text{ult.}}$ as was the case for the aluminum plate analysis. The reasons for this discrepancy have been discussed in the previous section.

(3) Summary:

The experimental measurement of dynamic strain during impact loading has been accomplished. The frequency



data of elastic wave propagation that has been obtained may be our most valuable tool for determining the value of the dynamic modulus of elasticity. The ability to measure the residual plastic strain component in a total strain level has also been demonstrated. This ability will allow us to very accurately determine the dynamic σ - ϵ curve from a number of tests. Information well beyond the empirical yield point $0.2\% \epsilon$ seems very possible. The preliminary magnitudes that have been recorded are satisfactory at this time. Agreement was good and it is felt that the experimental data would correlate very well with the predictions that could be made from the detailed analysis with proper boundary condition modification. The interplay of: (1) strain-gage reinforcement, (2) amplification factor based on pulse shape as well as frequency, (3) viscous damping of the porous material, (4) accuracy of E value, and (5) the compressional strain-gage behavior, have been briefly described. With an accurate experimental determination of E, σ may be accurately calculated from Equation (12) if we have made the analytical modifications indicated. Then we are confident that the dynamic amplification could be accurately determined for future use.



D. IMPACT TESTING OF Ni-Cd CELLS

1. Failure Criteria

True "failure" may be electrochemically defined as the cell's inability to deliver suitable discharge capacity following an impact environment. Since we have monitored a cell's open circuit voltage during impact, we can use this as an indicator of electrical damage. Typical of the information provided by this failure indicator are the phenomenon shown in Figure III-16. The upper traces display a cell's open circuit voltage from 5 msec. before impact to 15 msec after impact. The three oscillographs presented indicate the classifications of impact-induced damage:

(a) no cell damage	(N) $V_2 = V_1$	no ΔV @ impact
(b) questionable damage	(P) $V_2 = V_1$	ΔV @ impact (momentary shorting)
(c) failure	(f) $V_2 < V_1$	ΔV @ impact (complete shorting or loss of plates)

The type of behavior exhibited in (b) is of very questionable damage electrochemically. We assume that such a momentary short will have little effect on the following discharge capacity and certainly not prevent the cell from subsequent cycling. However, we will distinguish this observed behavior from that of a complete satisfactory performance.

UPPER BEAM:

OC VOLTAGE OF CELL

VERT: 1 VOLT/CM

HORIZ: 2 msec/CM

a)

NO CELL DAMAGE



b)

MOMENTARY
SHORTING
DURING IMPACT



c)

COMPLETE
CELL FAILURE
AT IMPACT



LOWER BEAM:

ACCELERATION

VERT: 1000g/CM

HORIZ: 0.5 msec/CM

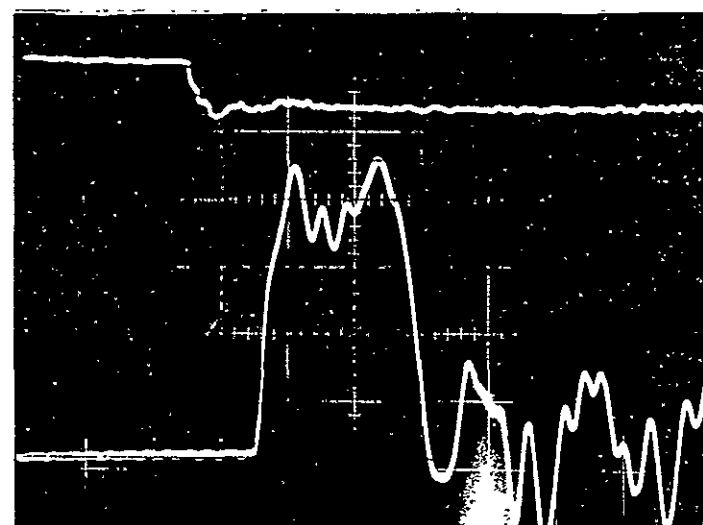
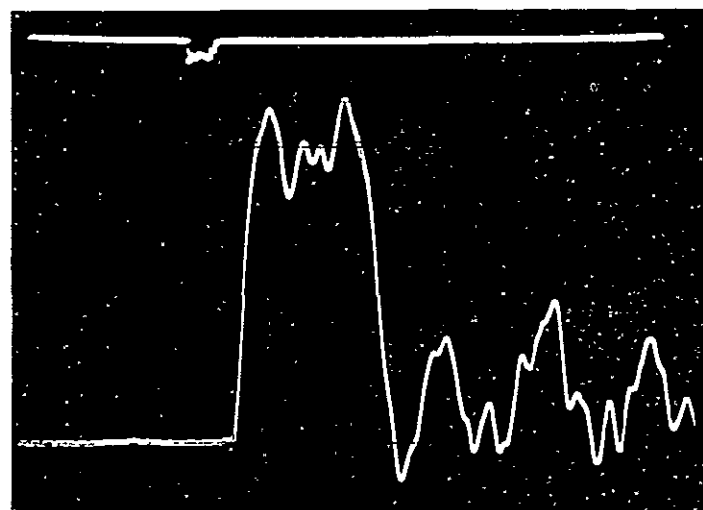
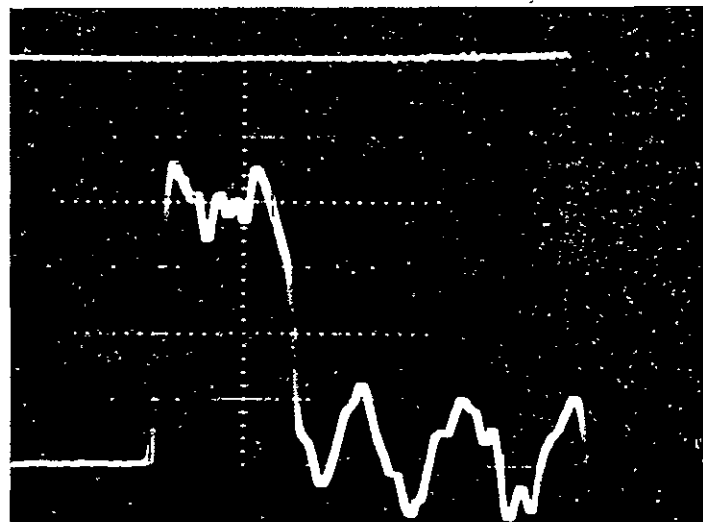


Figure III - 16

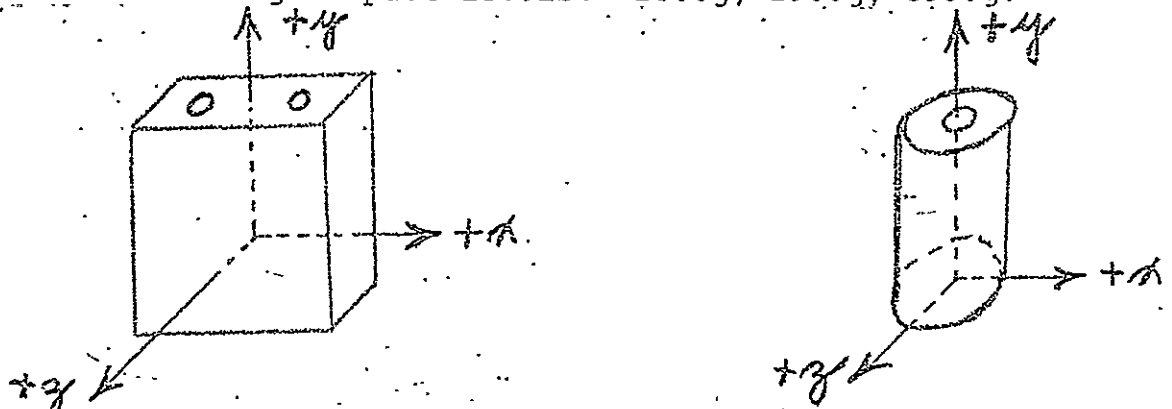


2. Existing Ni-Cd Cells (<5AH)

The total study of existing cells involved 52 impact tests to evaluate the performance of four (4) types of Ni-Cd cells:

- (1) C Sub-C without spindle (Part #006200)
- (2) CS Sub-C with spindle (Part #003800)
- (3) LC Long Sub-C without spindle (Part #005500)
- (4) P Prismatic 4 AH (Part #003500)

Each type of cell was tested in each non-redundant direction at three average impact levels: 2300g, 2960g, 3950g.



NOTE: Impact in +y direction implies:

+v in +y direction

+a in -y direction

impact surface -- terminal end (+y)

Each cell was fully characterized physically and electrochemically before impact testing. The cell was in the charged state and voltage was monitored during impact.

Following impact and discharge, each cell is further examined physically and electrochemically (post-



characterization) before complete failure analyses.

Table III D-1 summarizes the results of the complete study. Each type of cell is shown with the directions of impact included. The actual impact (meas. mean g-level and velocity) is stated and the observed cell behavior. It must be emphasized that none of these cells was designed specifically for impact-resistance. The inherent impact resistance exhibited (or the lack of it) is not meant to imply the actual shock resistance that can be attained by any of the configurations that have been studied. The obvious example is the prismatic cell where great gains could be expected for the $\pm y$ direction with a minimum of design improvement.

In general, we were encouraged by the preliminary analysis of this study. The cylindrical cells performed quite well, however, the Sub-C with spindle seems to have more potential damage in the x direction than the same cell configuration without a spindle. The long Sub-C seems to have experienced more difficulty in the $-y$ direction than in the $\pm y$ direction. Similarly, the prismatic results in the $\pm y$ directions were anticipated, but not the relatively poor behavior in the z direction at the highest impact level.

3. Testing of High-Capacity Cells (25 AH)

In addition to the programmed testing of smaller production



TABLE III D-1

Cell Type & Impact Direction		Low Impact Level (V) (g) ft/sec.			Med. Impact Level (V) (g) ft/sec.			High Impact Level (V) (g) ft/sec.		
		Beha- vior			Beha- vior			Beha- vior		
Sub-C	x	2000	(87)	N	2800	(106)	N	3800	(118)	N
Without	-y	2350	(86)	N	2700	(107)	N	4170	(122)	P
Spindle	+y	2350	(--)	N	3400	(102)	N	4100	(113)	N
								3500	(120)	P
								4200	(115)	N
Sub-C	x	2000	(88)	N	3400	(106)	N	3600	(122)	F
With	-y	2520	(89)	N	2750	(106)	P	3600	(123)	P
Spindle	+y	2050	(86)	N				4600	(121)	P
		2400	(87)	N	2800	(108)	P			
								4800	(119)	N
Long	z	2100	(89)	N	2700	(107)	N	3500	(122)	N
Sub-C	x	2350	(88)	N	3400	(106)	N	3500	(121)	N
Without	-y	2450	(86)	P	3120	(105)	P	3700	(119)	P
Spindle	+y	2300	(90)	N	3000	(105)	P			
					2700	(106)	N	3700	(117)	P
								4600	(116)	P
Prismatic	z	2250	(89)	N	2500	(104)	N	4070	(114)	P
4 A.H.	-							4020	(111)	P
	x	2300	(89)	N	2850	(105)	P	3800	(120)	F
	-y	2300	(88)	F	2800	(104)	F	4400	(120)	P
	+y	2600	(87)	F	3200	(102)	F	3100	(112)	F
								3800	(113)	F

CELL BEHAVIOR CODE: N - no damage
P - possible damage
F - failure



cells, 25 AH cylindrical prototype cells have been tested. It must be emphasized that these cells were heat-sterilizable but not designed for impact resistance. The tests were meant solely to study the design techniques and practical aspects that might occur with the larger cells.

a) Description of Large Capacity Cells

Since it was decided that a cell specifically designed for impact resistance would not be manufactured under the current contract, we felt it would be of value to study the large heat-sterilized Ni-Cd cells. Three cells of the approximately 25 AH size were available. Again, it must be emphasized that these cells were not designed for impact resistance but were of geometric value in supporting some of our data on plate strength.

Cell #PR38

(12) Positive Plates 6.6" lg. x 2.8" wd. x .0255" tk.
1.46 gm/in², 0.0596#/plate

(13) Negative Plates 6.6" lg. x 2.8" wd. x .026" tk.
1.61 gm/in², 0.0656#/plate

(26) Separator 0.005" tk.

Case 7-5/8" high x 2.925" wide x 0.832" tk. internal

Spacer 0.5" @ top

free SPACE:

Thickness --

Approx. 0.002"/plate

Width --

Approx. 1/16" each side

metallurgical
materials division



Height --

Approx. 7/16" Top

Approx. 1/16" Bottom

$$\text{from: } \sigma = \frac{F}{A} = \frac{W(N)}{w \cdot t}$$

$$4000 \text{ g} \Rightarrow \sigma_{\text{max pos}} = 3,340 \text{ psi}$$

(assume no
dynamic amplification) $\sigma_{\text{max neg}} = 3,610 \text{ psi}$

Cell #PR114

(10) Positive Plates 5.4"lg. x 2.8"wd. x 0.030" tk.
1.67 g/in² 0.0557 #/plate

(11) Negative Plates 5.4"lg. x 2.8"wd. x 0.032" tk.
2.04 g/in², 0.0680 #/plate

(22) Separator 0.005" tk.

Case 6.31"high x 2.925" wd. x 0.832" tk. internal

Spacer 0.5" tk. @ Top

free SPACE:

Thickness -- Approx. 0.0018"/Plate

Width -- Approx. 1/16" each side

Height -- Approx. 1/4" @ Top
Approx. 1/8" @ Bottom

$$\text{from: } \sigma = \frac{W(N)}{w \cdot t} \quad (\text{assume no dynamic ampl.})$$

$$4000 \text{ g} \Rightarrow \sigma_{\text{max pos}} = 2,650 \text{ psi}$$

$$\sigma_{\text{max pos}} = 3,040 \text{ psi}$$

Cell #C33

(3) Positive Plates 18"lg. x 4.2"wd. x 0.0255" tk.
1.465 g/in², 0.244 #/Plate



(4) Negative Plates 18"lg. x 4.2"wd. x 0.026"tk.
1.61 g/in², 0.268 #/Plate

(6) Separator 0.005" tk.

Case 2.931" I.D. x 5.532" int. height

Spacer 0.5" tk. @ Top

free SPACE:

Thickness -- Approx. 0.01"/Plate

Height -- Approx. 11/16"-13/16" @ Top

Approx. 3/16"-1/4" @ Bottom

from: $\sigma = \frac{W(N)}{w \cdot t}$ (assume no dyn. impl.)

$$4000 \text{ g} \Rightarrow \sigma_{\text{max pos}} = 2120 \text{ psi}$$

$$\sigma_{\text{max pos}} = 2290 \text{ psi}$$

One cell of each configuration had been previously cycled and only cell #PR114 was shorted prior to our testing. Initial indications were that the plates in all these cells would support the loading imposed by a 4000 g - 1 msec impact! The stresses that have been calculated do not exceed the values obtained during static bearing tests for plates with the same degree of lateral restraint. Dynamic bearing stress values gave excellent correspondence with the static values and indicated that a significant dynamic amplification factor is not required. The plates used in these cells are much longer than those used in static bearing tests and approximately equal in length to those used in dynamic bearing tests. This implies a lower natural

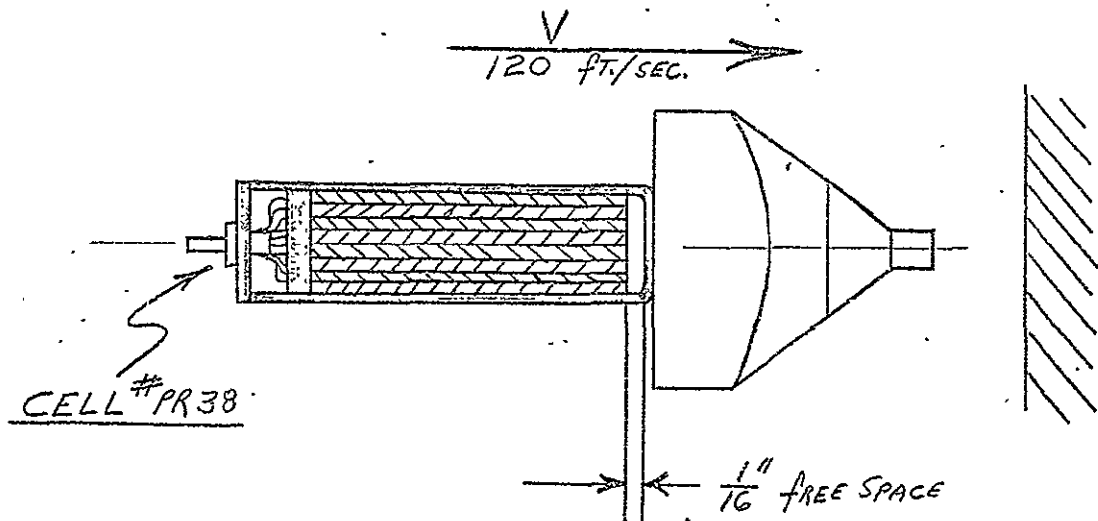


frequency and therefore even less concern for requiring a dynamic amplification factor. Thus our only conclusion from previous evaluation of the mechanical properties of Ni-Cd electrodes would be that the plates in any of these cells would survive impact.

b) Analysis of the Impact Process:

The preliminary conclusions are indeed founded on design data. However, we must examine the manner of loading. There is reason for concern over the amount of free space permitted within these cells. It will be shown that this is the main reason for emphasizing that these cells were not designed for impact resistance.

It is instructive to ask: Is a mere 1/16" vertical free space dangerous to a large cell's capability for impact resistance?





When impact occurs, the carriage, cell, and individual electrodes experience a 4000g gravitational field for 1 msec. The resultant force exerted on a negative electrode is:

$$F = ma = .262\#$$

This force will accelerate the electrode to some higher velocity which will be limited by the distance (and thus time) over which this acceleration force can act:

$$S = v_i t + 1/2 a t^2$$

$$t = .35\mu \text{ sec}$$

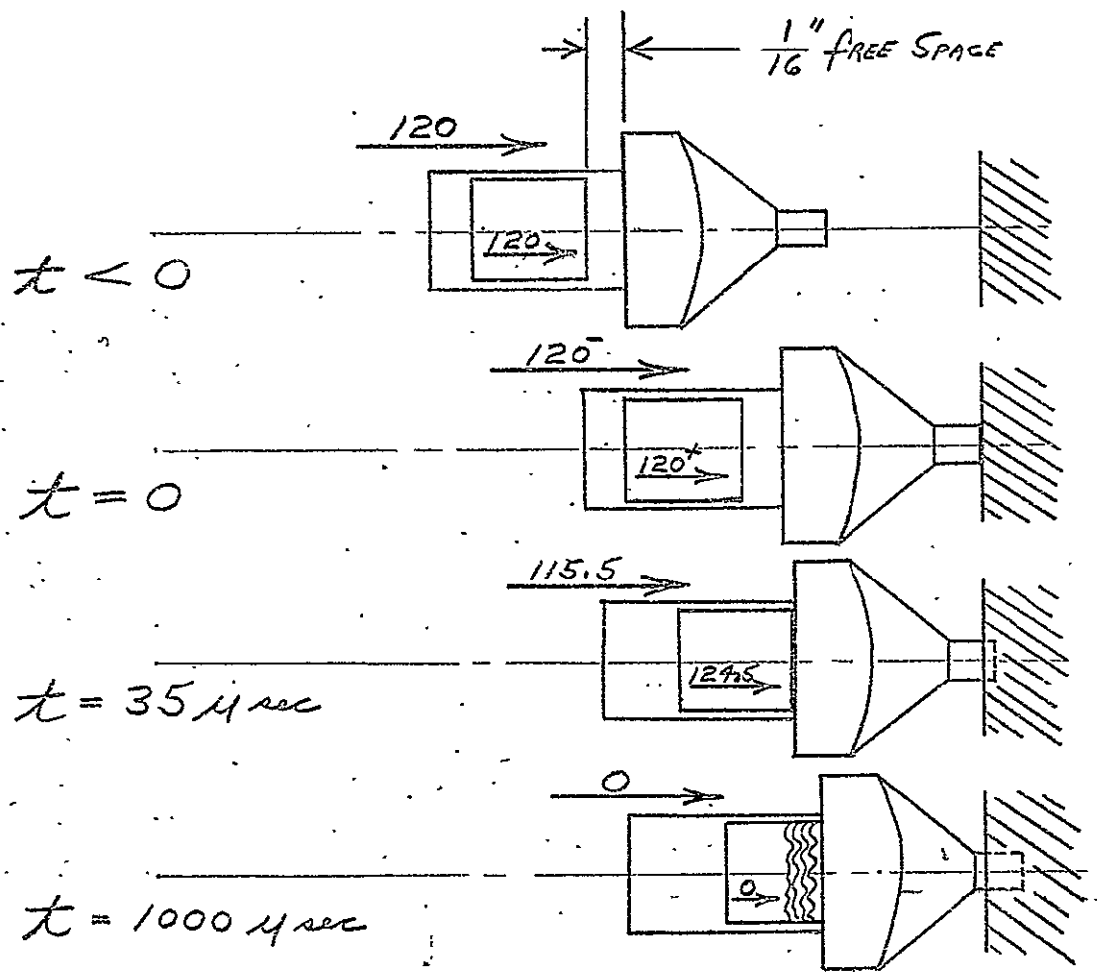
With this time available to cover the 1/16" free space, the final electrode velocity at impact will be:

$$v_f = v_i + at = 124.5 \text{ ft/sec}$$

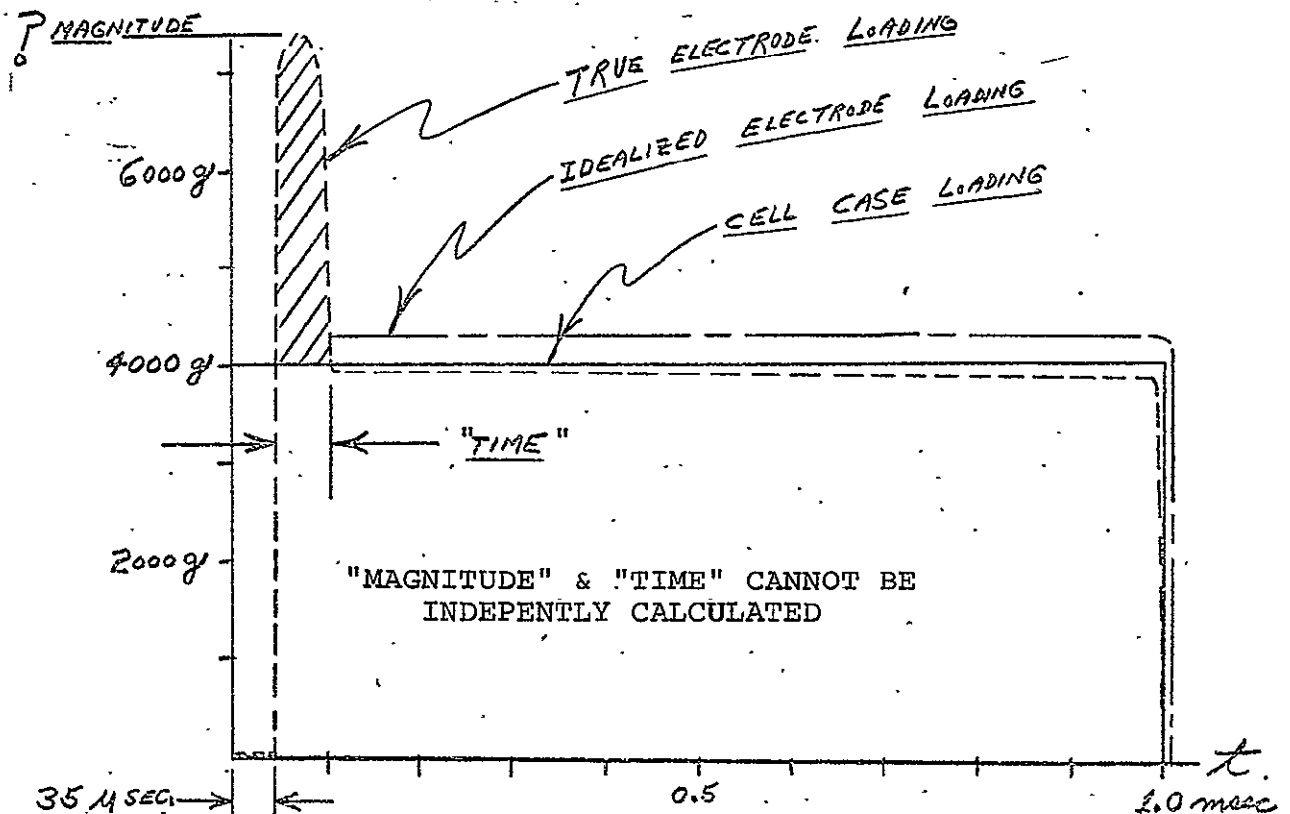
Now during the same time interval, the carriage and cell case have to be decelerated to a velocity of:

$$v_f = v_i + at = 115.5 \text{ ft/sec}$$

Diagrammatically we have:



In reality therefore, we have the carriage and cell case experiencing entirely different loading than the electrodes:





The cell case of course experiences the uniform deceleration loading. The electrode, however, has a relative velocity at its impact of 9 ft/sec which must be diminished before movement with the carriage (and 4000 g loading) can occur. Rapid deformation of the electrode will occur at this point, as loading increases for a short time. We can calculate the energy involved.

$$\int F \cdot dt = m (\Delta v) = (9 \text{ ft/sec})$$

or the shaded area of the spike. Unfortunately, we cannot determine the "magnitude" or "time" involved without additional experimental information describing the deformation process.

We can only assume that this is a small amount of energy and question if the electrode can accommodate it by deformation at maximum load for the remainder of time available. If the static bearing values are valid, the maximum stress that a negative electrode can support at .002"/plate lateral restraint is 3750 psi. Thus the maximum force is:

$$F = \sigma (w \cdot t) = 274\#$$

The amount of time required (by uniform force) to accomplish the total velocity change from 124.5 ft/sec \rightarrow 0 ft/sec is:

$$F \cdot t = m (\Delta V)$$

$$t = 0.926 \text{ msec}$$



This amount of time is < 0.965 msec still available and implies the deceleration could be accomplished safely (with a margin of only 39μ sec!).

This exercise was performed for a specific plate (2.36 length: width ratio) and assumed a mere $1/16$ " free space; yet, it dramatically illustrates the very fine distinction between failure and survival. Several points regarding this analysis must be noted:

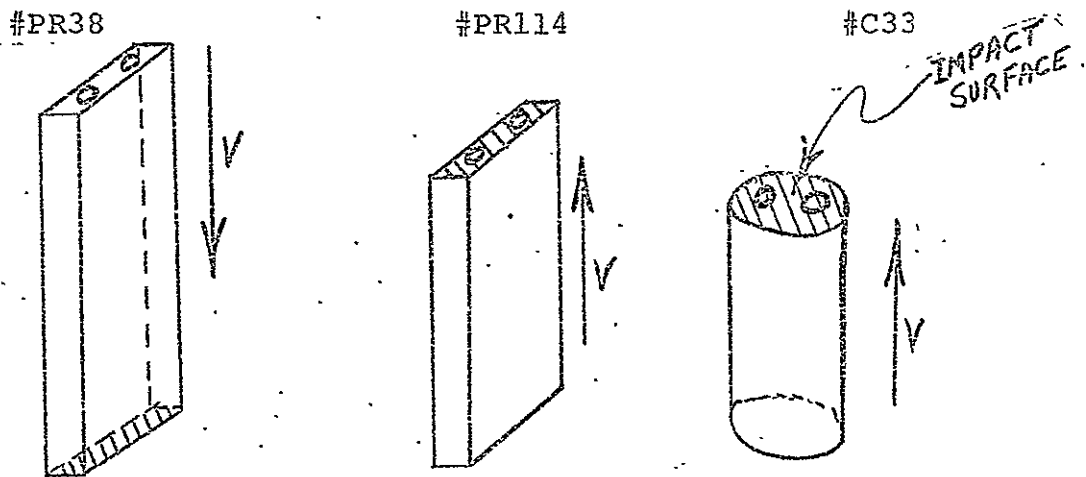
1. The additional weight of electrolyte saturation was not included in the mass of the electrode.
2. The total area of the end of the electrode was presumed to be available for supporting load.
3. Prior to initial impact, the electrode pack was presumed stationary -- maintaining its $1/16$ " relative distance to the cell case impact surface.
4. Most importantly, the effect of a damaging peak force at electrode impact was neglected in our final statement of apparent survival.

c) Results of Testing:

We shall now look at the results of impact tests performed on these cells and see that failure to achieve the above conditions will have rather obvious consequences.



The orientation of each cell during impact is shown below:



The description of the impact parameters for each cell (velocity, g-level, time) is given in Figure III-17 (a), (b), (c). The time history of open circuit voltage during impact is also shown. All three cells failed under the imposed impact. X-rays taken of each cell immediately after impact are shown in Figures III-18, III-19, III-20. Upon disassembly of each cell, photographs were taken of the electrode deformation and are shown in Figures III-21, III-22, III-23. The visual evidence of failure depicts quite vividly the kinds of deformation that can occur when some of the earlier design points are not realized. We may discuss these tests individually:

Cell #PR38:

The large prismatic cell failed in buckling of the electrodes. It was our decision to test each cell as

UPPER TRACE: 2 msec/cm

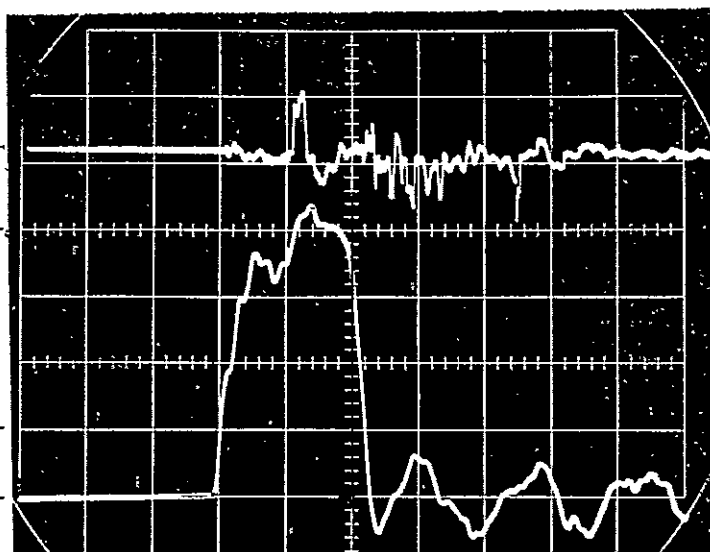
LOWER TRACE: 0.5 msec/cm

(a) CELL #PR 38

V = 105 ft/sec

1.2 VOLT

1000 g

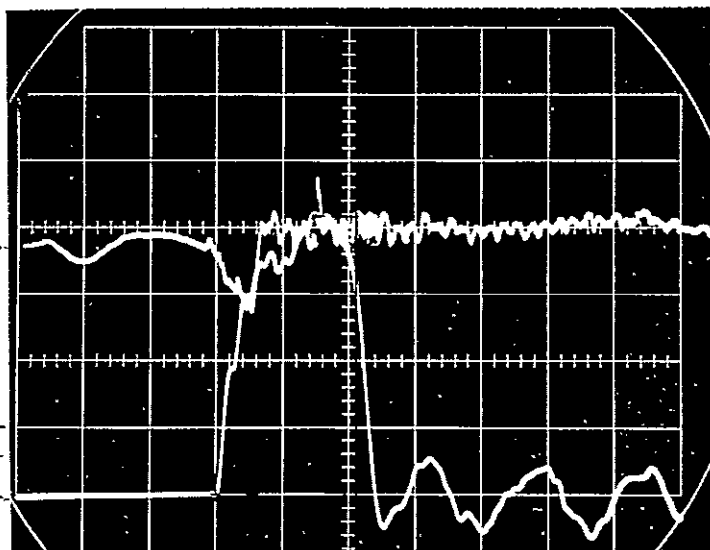


(b) CELL #PR 114

V = 104 ft/sec

EXTRANEEOUS SIGNAL
NOT CONNECTED
TO SHORTED CELL

1000 g



(c) CELL #C 33

V = 104 ft/sec

0.7 VOLT

1000 g

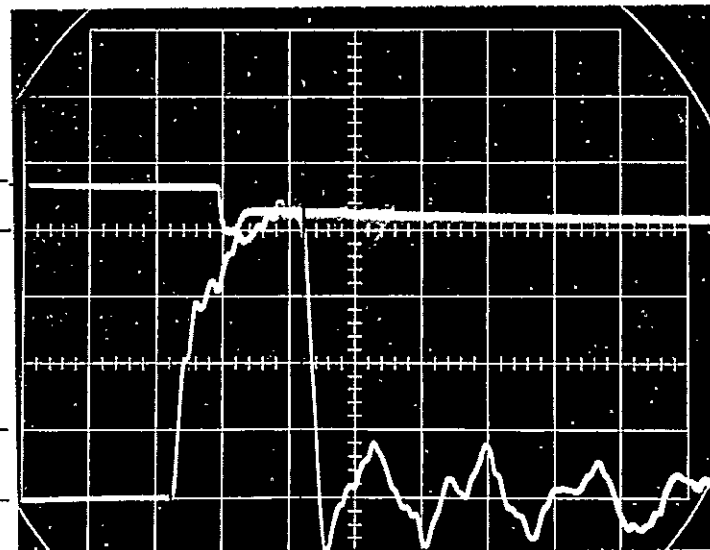
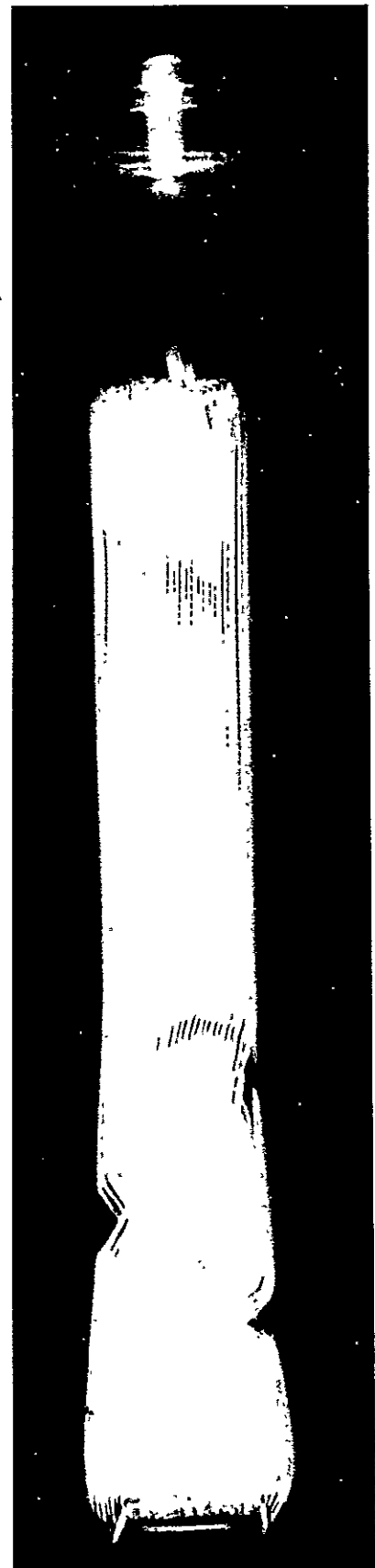
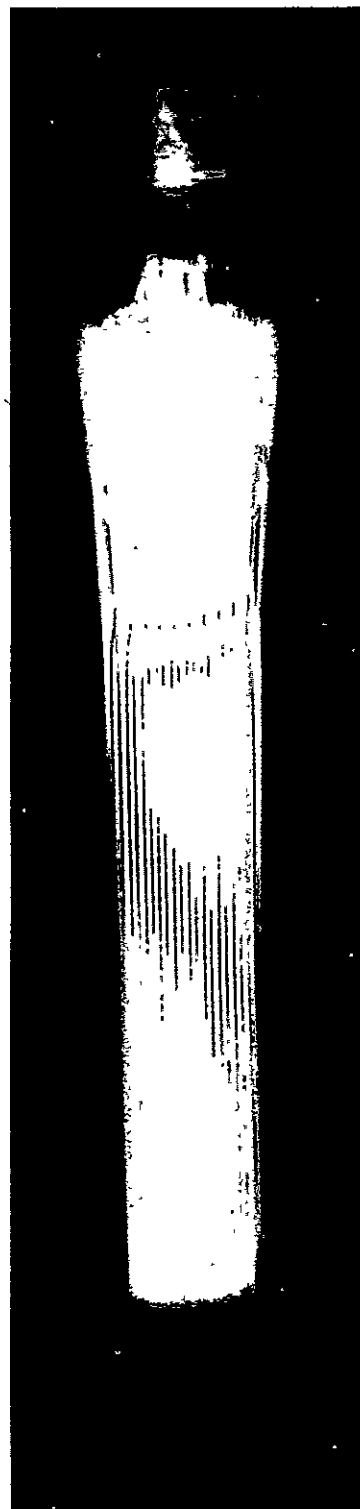


Figure III - 18



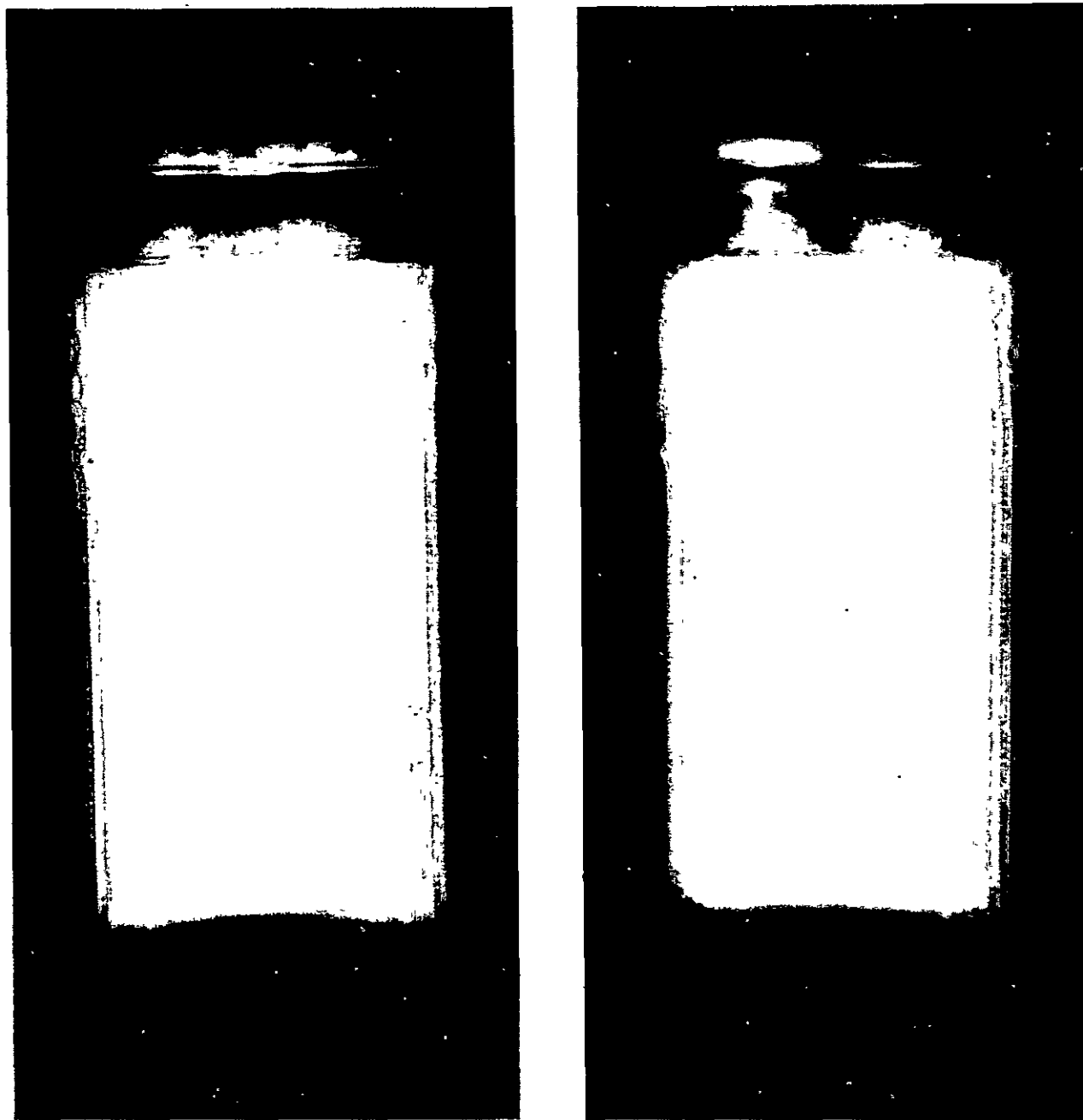
CELL #PR 38 POST-IMPACT

Figure III - 19



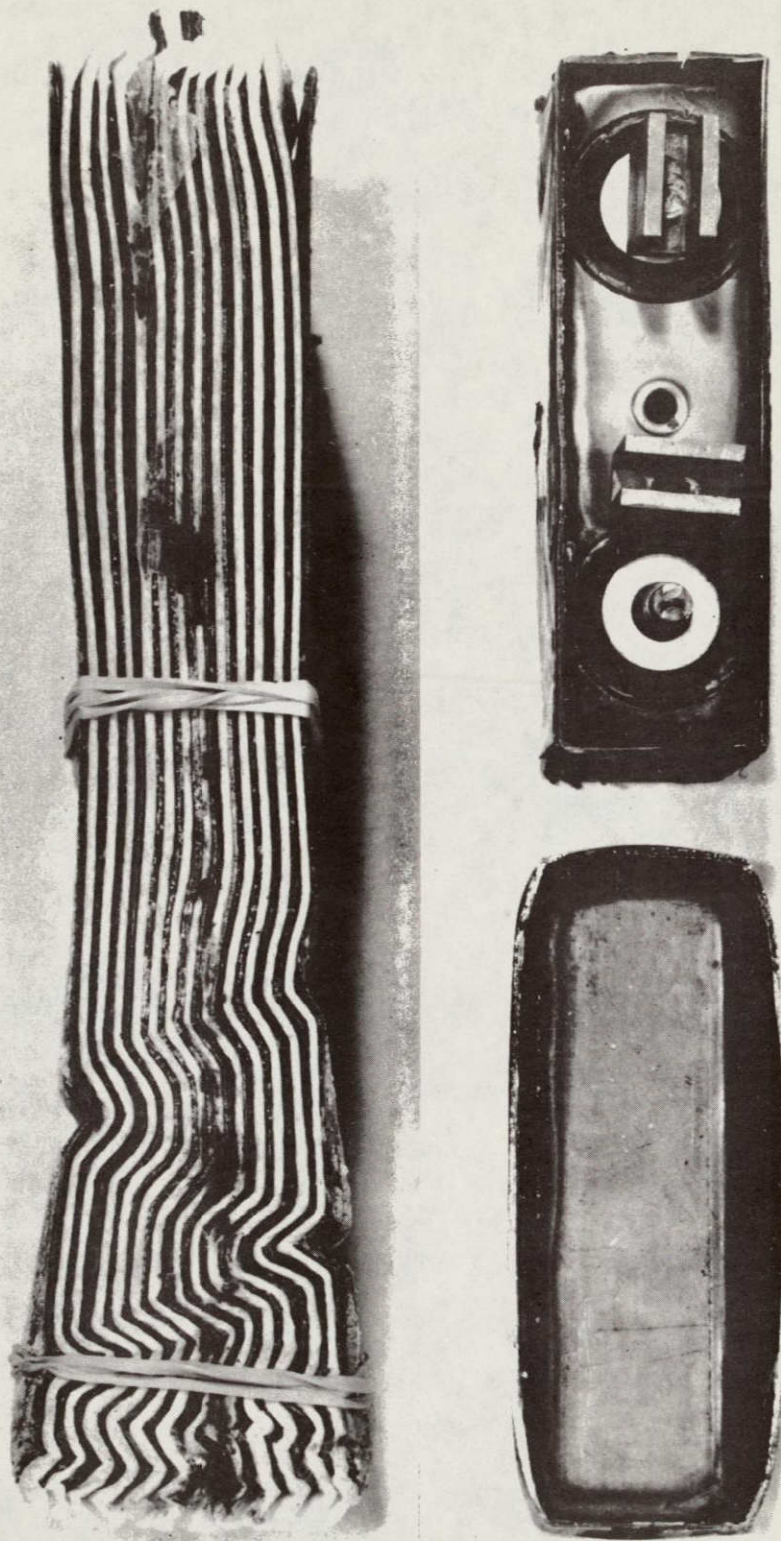
CELL #PR 114 POST-IMPACT

Figure III - 20



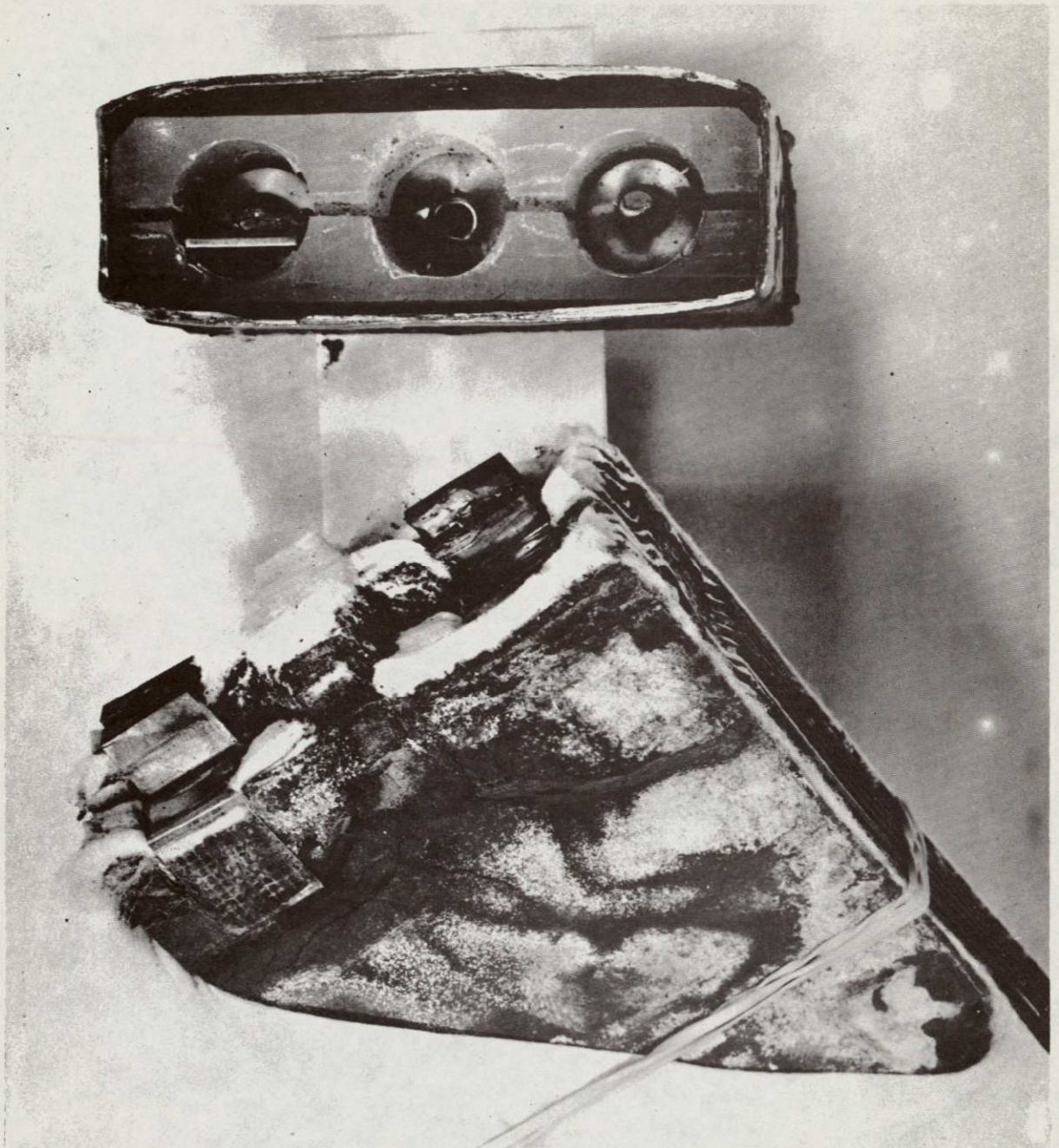
CELL #C 33 POST-IMPACT

Figure III - 21



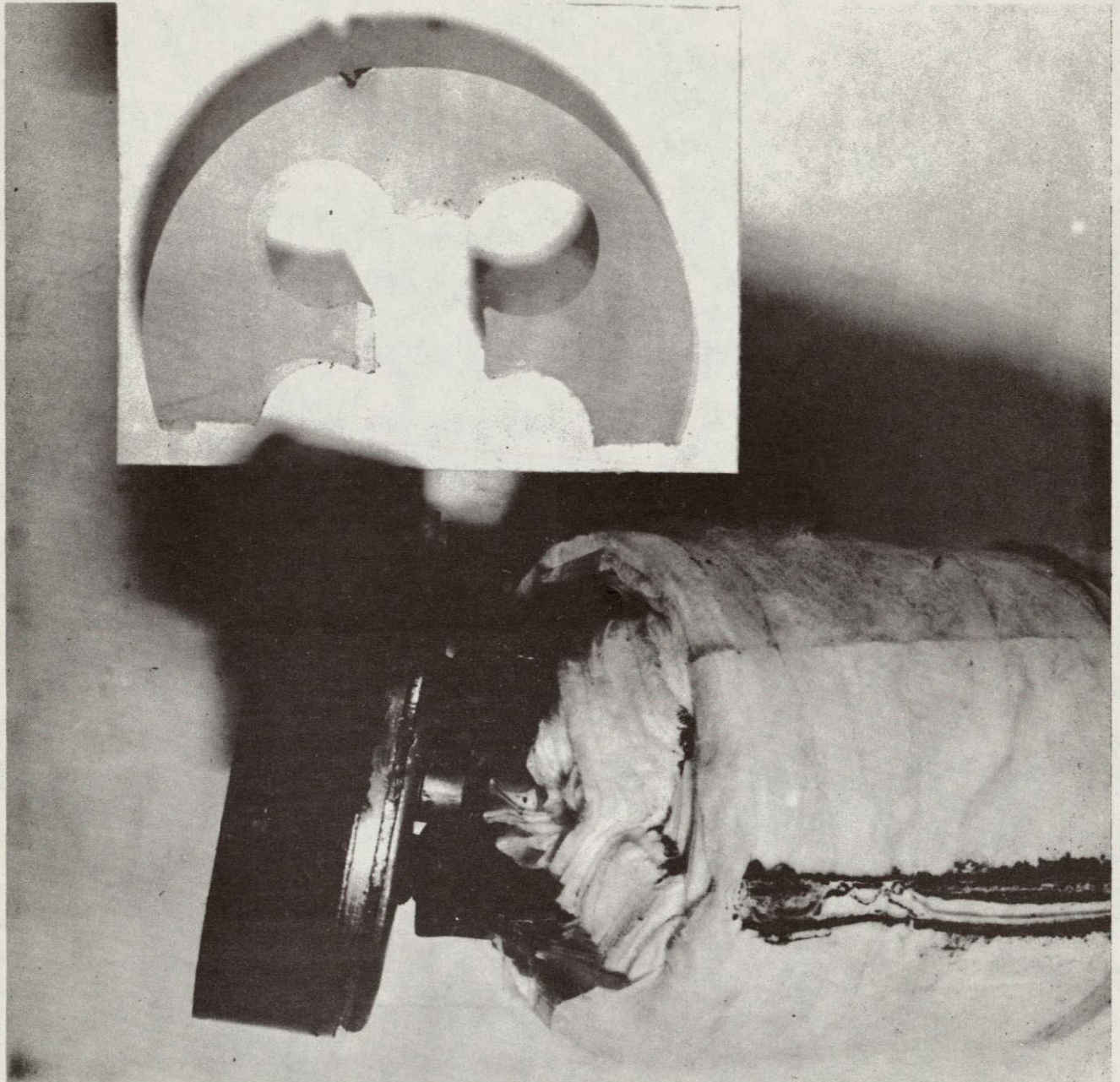
CELL #PR 38

Figure III - 22



CELL #PR 114

Figure III - 23



CELL #C 33



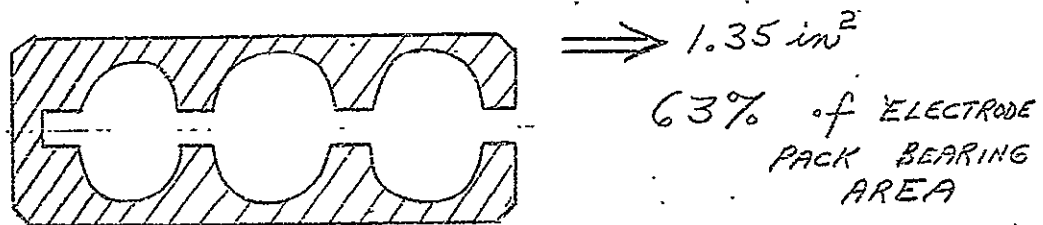
an entity in the sense that external restraints would not be used. This lack of restraint showed that the cell case alone is not of sufficient strength to support its own weight under inertial load and the lateral pressure imposed on it by the deforming electrode pack. The electrode pack apparently had sufficient lateral restraint and support from its terminals to withstand the loading during acceleration. This is both a visual observation and the presence of full voltage at impact. Under deceleration a much lower degree of lateral restraint was available from the cell case, and the electrodes had little difficulty in buckling (Figure III-18 & 21). The electrodes could not reach the full value of stress which they are capable of supporting. The post-mortem failure analysis also showed that one terminal collector had apparently not been welded. This is the most likely cause for the erratic voltage trace shown in Figure III-17(a). We note, however, that voltage was maintained on this cell despite the defective terminal and the extensive deformation. It is our contention that this cell would not have experienced failure if thicker axial spacers and lateral restraining plates had been used.

Cell #PR114:

The smaller prismatic cell had the largest free space



at the impact end of any cell and this is largely responsible for the damage. We cannot make a judgment on the motions during acceleration. Although there is no visual damage to the bottom of the electrode pack, we do not have a trace to indicate the voltage at impact. During deceleration both terminal collectors were fractured. The most significant lesson in the study of this cell is the total ineffectiveness of the spacer design.



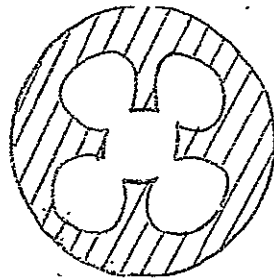
The predicted stresses for this cell become meaningless under this situation and failure is the obvious consequence. The visual observations of the top of the electrode pack dramatically show the areas which supported load (Figures III-19 & 22). The spacer vent hole area is quite emphatic. We note that the failure mode in supported areas is again buckling. This again indicates that low lateral restraint (static plate spacing and dynamic cell case expansion) prevented the electrodes from reaching their ultimate compression stress.

Cell #C33:

The cylindrical cell developed direct shorting in the



terminal areas of the cell (Figure III-17(c)) as electrode motion was severe during impact. Again, the spacer design was solely responsible for this failure.



⇒ 2.60 in²
59% of electrode pack
bearing area

As shown in Figure III-20 and Figure III-23, the plates moved forward into the open areas of the spacer. The open core in particular had large axial excursion. Again, there was no damage to the cell during acceleration (visual observation and cell voltage trace). It is significant that this cylindrical configuration has resisted nearly totally the buckling mode of failure. Even in the areas where severe bearing loads were possible, the x-ray analysis shows only traces of buckling. The stability of the cylindrical cell is apparent mainly because of the inherent cell case design. What we have seen is that the cell cases in the two prismatic designs (without the benefit of restraining plates as normally used on batteries) were the prime cause of lowering lateral restraint under impact conditions. This confirms our contention that our electrodes themselves will



resist buckling and go on to the true ultimate compressional stress that we have established.

In conclusion, the study of these cells has provided the design emphasis that we desired. The importance of static assembly plate spacing, the use of spacers that maximize effective bearing area and axial thickness, and the cell case contribution to pack integrity have been adequately demonstrated. In short, we have stated the precautions that must precede the use of our design data to predict a cell's behavior under impact loading.



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